# POWER

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# Cash in Your Ideas

This is a call for help.

We want a greater variety of topics and treatment in the matter on this page, than is possible from the limited number on our own staff.

To enlist outside aid we make the accompanying offer for all accepted material.

Cartoons are particularly invited. They call for little or no text, telling their story by lines rather than words.

**$5 for an idea for a foreword or a cartoon.**
**$10 for a rough sketch for a cartoon.**
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Any material inappropriate for foreword use but suitable elsewhere in the paper, will be so used and paid for at our regular rates.

Matter will be paid for upon acceptance, or if unavailable, promptly returned.

We will not suggest as to selection of subjects, for fear of defeating our purpose; namely, to get new views into these pages, not limited by our own range of vision.

Take past forewords for a guide, as to *what* and *what not* to send. The general character or style of treatment may be used as a pattern, but the same topics should be avoided, unless you can treat them from a new angle or bring out some new idea.

What interests you, as a reader of POWER; may be expected to interest others. Keep to lines related to our field, but not technical things only. The human side should come in for its share, but don't always preach a sermon.

Just think of a foreword as an informal editorial; the rest may be left to your judgement. Above all don't be afraid to make a try. Perhaps you underestimate your own ability. We are ready to assist by touching up here or there if necessary.

Now, let's hear from you.

# Station of San Diego Electric Ry. Co.

By Leroy W. Allison

**A 4200-kw. plant employing a high-pressure reciprocating engine and low-pressure turbines. The latter drive their generators through Melville-MacAlpine reduction gears.**

To meet the increased and growing demands of its city and interurban transportation system, the San Diego Electric Ry. Co., of San Diego, Calif., has erected a new electric-generating station at Arctic and E Sts. The plant has a capacity of 4200 kw. and is employed exclusively for railway service. In general construction and equipment it offers the highest type of the present-day generating plant and no expense has been spared to effect a commendatory result to the owners, the builders and the city. The electric traction system now comprises about 56 miles of local street railway and 18.7 miles of interurban lines to Otay, Point Loma and Ocean Beach, with other long distance lines contemplated.

The generating station, about 10 years ago, located on the same site as the new structure, had a capacity of approximately 400 kw. This consisted of G. E. generators, belt-driven by a 250-hp. tandem Corliss engine and a 250-hp. automatic cutoff American Ball unit. Five years later one 1000-kw. and two 500-kw. turbo-generators, with three new 250-hp. Keeler boilers, were installed.

### Building

The former power house was demolished early in 1910 to make way for an entirely new plant, as shown in Fig. 1. The building, comprising boiler and engine rooms, is 160x170 ft. over all, with a basement under the latter room, 18 ft. high, for the auxiliary equipment. It is constructed of reinforced concrete throughout with steel trusses and roof framing. Corresponding with the imposing exterior appearance, the interior walls, columns and gallery (Fig. 2) form a noticeable contrast to customary construction. The balustrades surmounting the gallery are of solid concrete, cast from a prepared design.

The side wall of the building, adjacent to the low structure at the right of the illustration, Fig. 1, and which is now used as a paint shop by the company, is constructed in a temporary manner to allow for erecting an exact duplicate of the present plant on this adjoining property. The station will then be symmetrical about the large doorway at this point and will cover an entire city block. A standard-gage railway track enters the plant through this doorway and runs to the boiler room. The track, after crossing Arctic St., will connect with the new San Diego & Arizona Ry., a steam line now in course of construction and a subsidiary property of the Spreckels companies, which operate the electric-railway company.

### Boiler Room

The boiler room is 97x160 ft. in size and is approximately one-half the height of the engine room, as will be noticed from Fig. 1. The installation comprises nine water-tube boilers aggregating 2200 hp., consisting of four Babcock & Wilcox units of 200 hp. each, two Keeler boilers of 400 hp. each and three Keeler boilers of 200 hp. each. They are arranged in three batteries of two units each and three single, as shown in the plan, Fig. 3. The boilers take up about one-half of the room, the remaining area allowing for future additions. The furnaces have Leahy oil burners and operate at a working steam pressure of 180 lb. Two self-supporting steel stacks, 8 ft. in diameter, carry off the products of combustion.

The boiler-feed pumps and similar equipment are arranged along the wall separating the boiler and engine rooms. One 14x9x24-in. Blake vertical pump supplies the boiler feed, supplemented by two 10x5½x10-in. Worthington horizontal pumps. These are connected to the boilers by a 4-in. feed line. A 12x 8½x10-in. Worthington pump, working against a water pressure of 200 lb., is used for boiler washing and two 6x4x6-in. duplex pumps deliver the fuel oil at 40 lb. pressure.

An 8-in. suction line connects the boiler-feed pumps to the hotwell, and continues to a Day marine filter. The pumps are also connected by a 6-in. line to a 4000-hp. closed feed-water heater, which has a 12-in. exhaust to the atmosphere. A detail of this piping is given in Fig. 4.

### Engine Room

A view of the engine room is shown in Fig. 2 and a plan in Fig. 3. As in the boiler room, only about half of the available space is now utilized, the remainder allows for duplicating the present installation.

One 28x60x48-in. cross-compound Allis-Chalmers Corliss engine is centrally located, as shown, directly connected to a 1200-kw. direct-current Westinghouse generator. Steam is supplied through a 10-in. main, which branches from a 14-in. main header in the boiler room with connection to a 10-in. steam separator. The exhaust line is 22 in. in diameter, reducing to a 10-in. line as the supply to a low-pressure turbine. This is a 1000-kw. Westinghouse machine and drives its generator through a Melville-MacAlpine reduction gear. The unit may be seen opposite the Corliss engine in Fig. 2. It operates at 3600 r.p.m. on the turbine end, and at 514 r.p.m. on the generator end. The turbine takes exhaust steam from the reciprocating unit at 16 lb. pressure absolute, and the throttle is controlled by a 42-in. gate valve operated hydraulically in the basement. A board at the turbine end of the machine is

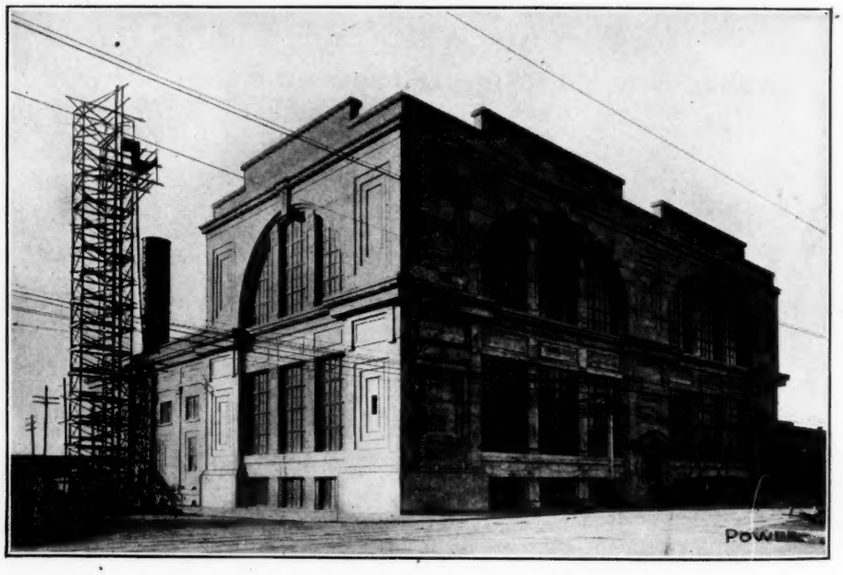

Fig. 1. Exterior View of Building

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equipped with gages for steam, oil, vacuum, gland or water pressure, and a speed indicator.

As indicated in the typical load curve, Fig. 5, the turbine carries approximately 98 per cent. of the engine load during a 24-hour period, at times running as high as 1200 amp. on the machine to 1000 amp. on the Corliss unit, at a pressure of 500 volts. The turbine has been in daily service since August, 1911, giving continuously efficient performance.

Another turbine unit of the same type and size is being installed alongside the first unit, as shown in Fig. 3. Former equipment will be used to effect this change, augmented by a reduction gear. The machine will also take steam from the Corliss engine.

ly interesting and has many original features.

As shown in Fig. 3. the system comprises two 60-in. lines of reinforced-concrete pipe, each 766 ft. long, with intake and terminal chambers. The two lines of pipe, in 3-ft. lengths, were laid on 7-ft. centers, affording a clearance of 12 in. The pipe was placed directly upon the sand, where the nature of the soil permitted, with the inside bottom 6 ft. below grade. For additional foundation as required, 3x3-in. mudsills of redwood were laid on 3-ft. centers, with two 8x8-in. stringers, beveled on one edge to support each line of pipe. Concrete manholes were placed at two street intersections along the route.

The intake chamber, near the bulkhead flanked with sheet piling, and 4x6-in. strips have been placed for a distance of 50 ft. out into the bay, as indicated, terminating about 15 ft. apart. This piling provides against the bypassing of any water from the discharge to the intake pipe. On the outer side of the dredged area, a wooden grill-work, similar to a picket fence, is built to keep out seaweed and other substances.

The terminal chamber is about 29 ft. from the power station, adjacent to E St. It is 18 ft. 3 in. by 20 ft. 3 in. inside and 24 ft. high. The footing is of the same size as the intake chamber and the exterior walls vary from 24 in. at the bottom to 12 in. wide at the top. The chamber is entirely covered on top, except for a manhole section, with a reinforced-concrete slab flooring. Openings similar to those in the intake chamber are provided for the entrance of the pipe lines and sluice gates are installed to operate from a wheel-stand on a platform 17 ft. above the floor of the chamber. The terminal chamber is connected to the plant by four tunnels.

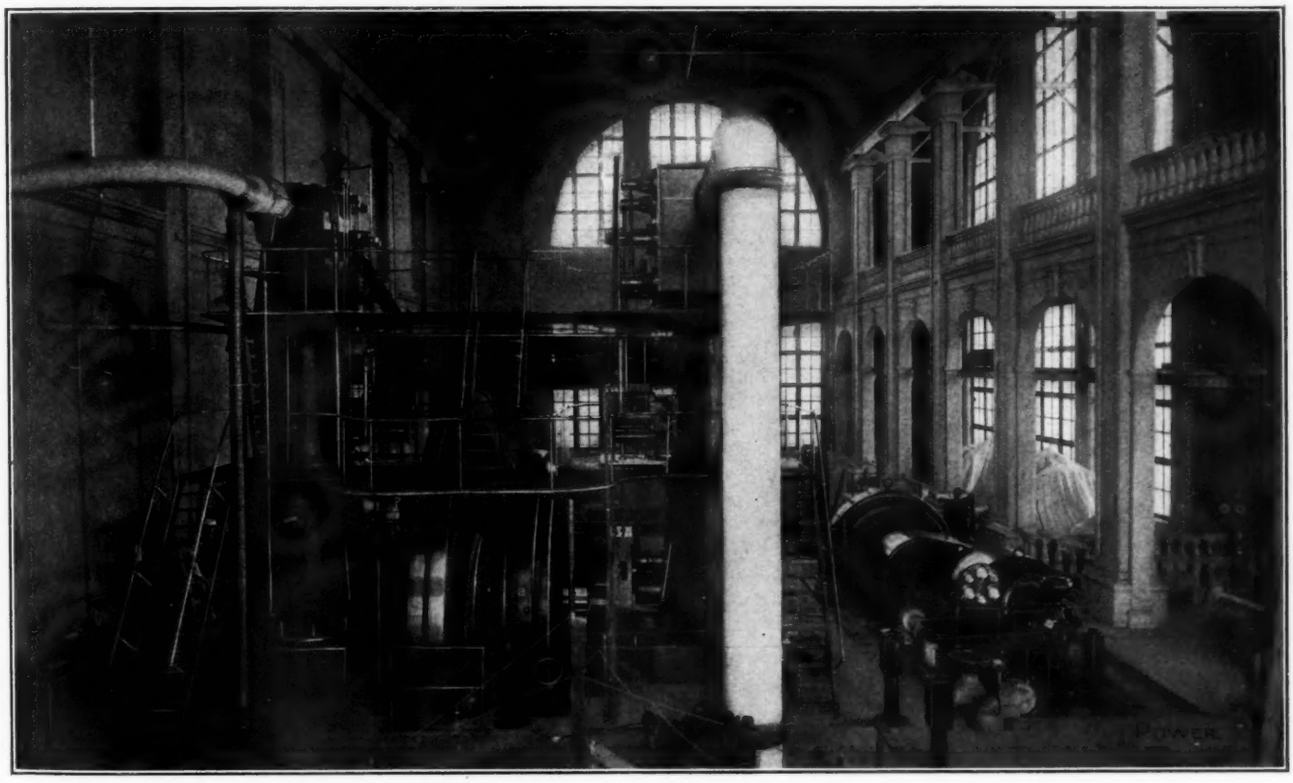


FIG. 2. VIEW OF ENGINE ROOM AND GALLERY

### AUXILIARY ENGINE-ROOM EQUIPMENT

As shown on the plan, two 500-kw. Westinghouse motor-generator sets are opposite the new turbine installation and two similar machines will later be located on the gallery floor. One 15-kw. G. E. horizontal steam-turbine lighting set is also installed.

A 65-ton Shaw electric traveling crane spans the engine-room main bay, while a 20-ton crane is installed beneath the gallery; both run the length of the building.

### SALT WATER CIRCULATING SYSTEM

Condensing water is taken from the Bay of San Diego, about 1000 ft. distant. The salt-water system is particular-

line at the foot of E St., is divided into two compartments, one for either line of pipe. These are 17 ft. high or approximately 2 ft. above the highest water line ever recorded at that point. The footing of the chamber is 24 in. thick; the interior walls vary in thickness from 24 in. at the base to 18 in. at the top, and are heavily reinforced with twisted rods. The partition wall is 12 in. thick and is similarly built. The pipes have cast-iron sluice-gate valves, operated from wheel-stands on top of the chamber. The chamber is covered on top with 2x12-in. planking and has an iron-pipe guard railing.

A channel 7 ft. wide and of the same depth has been dredged for each pipe line, beyond the intake chamber and

The entire system has been designed with a view to interchanging the service of the respective lines. Either line may be employed as intake or discharge. The terminal chamber has been constructed to effect this condition and also that either line may be closed temporarily for repairs, etc. The sluice gates additionally provide that the suction and discharge pipes from the plant enter the respective

sea line as desired. This alternation affords a means of keeping the line clear of seaweed and obstructions, and it is for this reason, primarily, that the concrete pipes are laid at the same level.

Through the tunnels between the terminal chamber and the power station, connection is made by two 20-in. suction lines, an 8-in. suction sump line and a 30-in. discharge line, all of cast-iron pipe. An additional 28-in. suction line is to be installed later.

The two 20-in. suction lines are operated by 26-in. Worthington horizontal centrifugal pumps, directly connected to a 125-hp. American Ball engine, both being in duplicate. The suction lines connect with three Wheeler surface condensers. The other auxiliaries include a 4-in. Worthington centrifugal pump directly connected to a 30-hp. Westinghouse motor; a pump of the same size, type and operation is used in connection with a hotwell adjoining the 5000-sq.ft. Wheeler condenser and permits its operation as an auxiliary to the condenser for the low-pressure turbine or Corliss unit.

An 18-ft. drop at high tide creates a siphon effect on the discharge end of the condenser, giving an approximate vacuum of 10 in., proportionately reducing the load on the engines driving the pumps. The circulating discharge water is maintained at 80 deg. temperature and is supplied to a neighboring bath house, as required, by an 8x8x10-in. pump.

### Electrical Equipment

The switchboards are temporarily located on the engine-room floor, as shown in Fig. 3, but soon will be moved to the

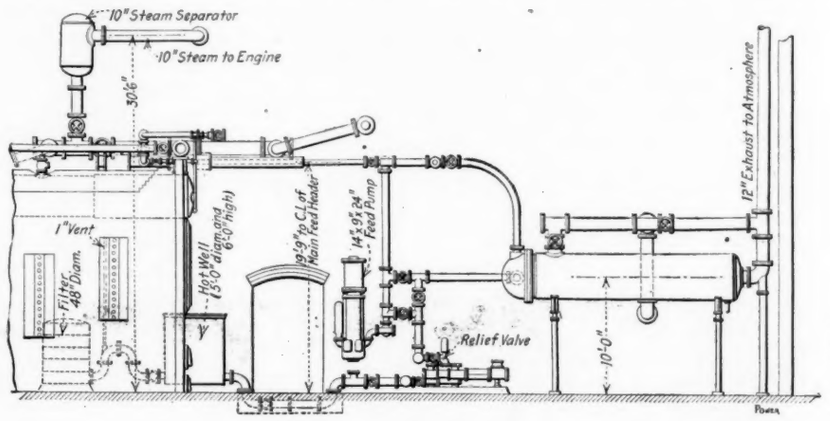



Fig. 4. Section through Boiler Room

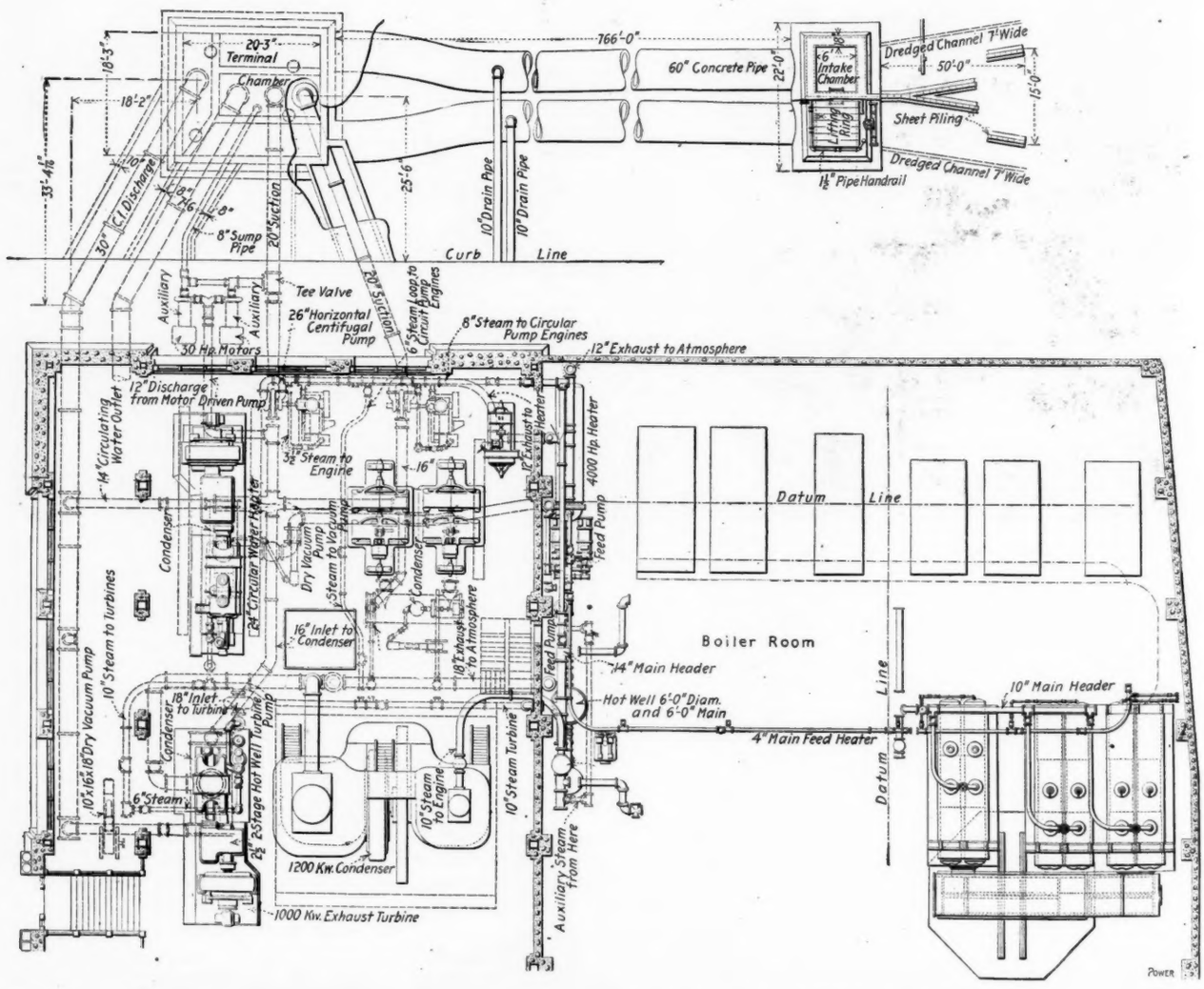



Fig. 3. General Layout of Plant Showing Engine and Boiler Rooms and Intake and Discharge Tunnels

gallery floor, which has been constructed primarily as a high-tension gallery.

The boards are of slate and comprise 19 panels in two sections. Sixteen of the panels are for direct-current service and the others for alternating current. Of the former, five are generator panels, eight, double-feeder panels, and three, single-feeder panels. On the alternating-current side, two are generator-feeder panels and the third a tie-in panel for outside service. The board is fully equipped with the usual electrical-measuring instruments of General Electric and Westinghouse type. Direct current at 500 volts is supplied for the railway service.

The outside leads, shown in Fig. 1, are temporary, and underground conduits leading from the basement, where the busbars and similar apparatus are located, are now in course of construction. To comply with a recent city ordinance the company has just completed the construction of underground conduits for its feeder wires in the business district.

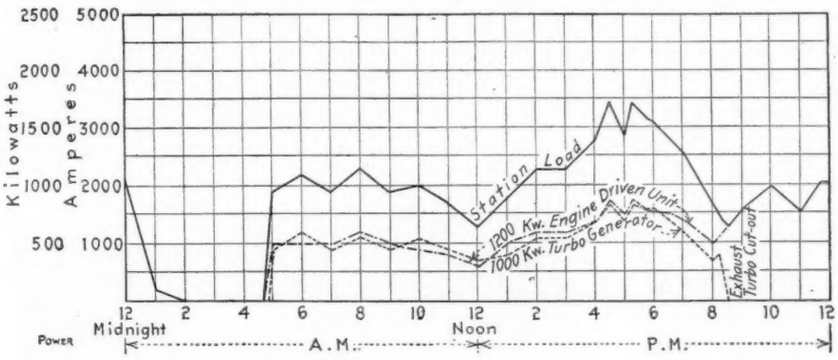



FIG. 5. TYPICAL LOAD CURVE

### STATION AUXILIARIES

The plant is equipped with a Richardson gravity-oiling system and a Bowser self-measuring oil-storage outfit is installed. A blowoff tank is in the basement, with a drain to the sewer, and a 9½x9½x12-in. air pump is used for cleaning the generators.

The company is at present constructing an underground fuel-oil storage tank on the opposite side of E St. This will be of reinforced concrete with a ¼-in. steel lining; it will have three compartments and a total capacity of 12,000 bbl. A pipe line will lead directly thereto from a private wharf of the company in the Bay of San Diego, permitting it to purchase its fuel from tank steamers at an exceptionally low rate. The oil-storage tank will connect with a day storage plant at the power station.

A city franchise has been granted the company to construct an underground heating line through the business section, principally on D St., where the Union Office building, the Spreckels Theaters and the Spreckels and Hanam Hotels, the property of the Spreckels Companies, are located. As soon as this line is built these buildings will be heated by exhaust steam from the power plant.

The station has cost approximately $350,000 and the future extensions contemplated will involve a considerable sum. Much credit is due the management for maintaining regular service throughout the entire reconstruction of the plant and causing no interruptions in the transportation system.

The building was designed by Andrew Ervast, chief engineer for the company, and was erected under his direct supervision. The power-plant design and the installation of machinery was carried on under the direction of Hunt, Mirk & Co., engineers and contractors, San Francisco.

## Variable-Speed Motor Drive for Stokers

The Detroit United Ry. has installed compound-wound variable-speed motor drives on all of the stokers in its main power houses, with the exception of those upon four boilers. The motors were put upon the opposite ends of the shafts from the steam engines, which were installed with the original apparatus, and the motors for two batteries, consisting of four boilers, require from 19 to 21 kw.-hr. daily.

The engines operate very nicely at fixed loads, but under certain conditions have to be slowed down so much that they get beyond the control of the governor. Then they are likely to stop when the mechanism passes the center and, unless the attendant notices it, the upper part of the grate becomes bare and incomplete combustion takes place. When the fireman comes back up the line and finds the engine stopped, he opens it wide, forcing in a lot of green coal, which produces smoke, etc. The motors under reduced-speed conditions operate very nicely.

## Engine Connected up Backward

BY J. H. STRATTON

Last summer I was asked by the superintendent of a neighboring company to see what I could do with his lighting plant, which consisted of a small dynamo direct-connected to a vertical engine, of the shaft-governed, piston-valve type, running at 650 r.p.m. He said that he had been unable to keep a man for more than a month and had to keep a hose turned on the crankpin every night and babbitt the brasses every two weeks, the engine sounding like a rock crusher.

We drove over to the plant, and when the engine was started I fully agreed with the superintendent that it sounded like a rock crusher.

I decided to try it for a few nights and while looking things over, the superintendent gave me the history of the plant, telling me that it had been installed by one of the best electricians in the oil fields, who, after trying for three days to get the lamps to light without any visible success, had left for parts unknown. The next man finally produced light and decided that the pounding was caused by the crankshaft being bent, so he took it out and sent it to a machine shop (10 m. away), telling them to straighten it. They placed it in a lathe and found it to be straight and true and sent it back with a note to that effect, but the engineer sent it to another shop to be straightened, as he said he knew "when a crankshaft was bent and when it was not"; the second shop came to the same conclusion and sent it back, whereupon the engineer got offended and quit.

The next man said the cause of the crankpin getting so hot was lumps on the pin, so one morning after shutting down he loosened the brasses and put in some powdered emery "to grind off the lumps" and then he quit by special request.

Several others had similar experiences with the engine during its 12 months of operation and finally it was up to me. I noticed that with a steady load the voltage would vary from 10 to 15 volts when the engine was running on a wide-open throttle. The superintendent told me that the governor never did work right, as the only way a steady voltage could be obtained was by cutting out all the rheostat resistance and running the engine on the throttle.

The first night I began exploring. The exhaust sounded like that from a 100-hp. gas engine. I noticed that the engine had a direct valve motion but the steam pipe was connected to the top of the steam chest. About this time I had discovered also that the steam and exhaust pipes were the same size (1½ in.). Following up this clew, I found that the

inlet to which the steam pipe was attached, was bushed from 2 to 1½ in. This told me at a glance where the trouble was; the steam pipe had been connected where the exhaust pipe should have been and the engine was running

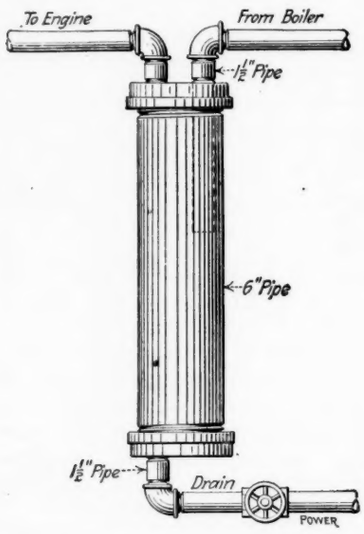



FIG. 1. DRAIN

backward, and had been doing so for 12 months.

The next morning I reported having located the source of the trouble and upon explaining it to the superintendent he laughed at me and said it was impossible, as an engine would not run with the pipes connected that way.

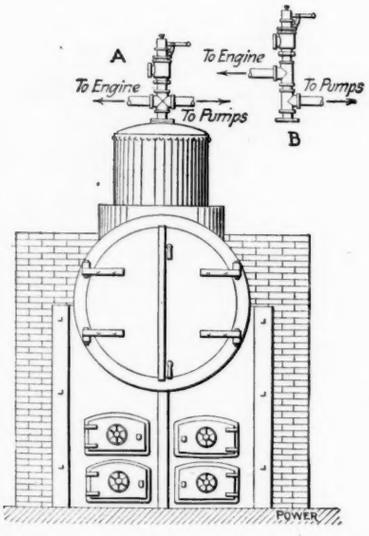



FIG. 2. ORIGINAL AND FINAL PIPE CONNECTIONS

I suggested that he walk down to the engine, whereupon I took a steam gage off one of the boilers and connected it to the drain valve of the cylinder. I knew that if the connections were wrong the gage would show more pressure on the cylinder than was on the boiler. Sure enough, when the engine was started the gage showed 180 lb., while there was only 120 lb. on the boiler and the pressure on the cylinder did not drop below 120 lb. The hand of the gage just traveled back and forth from 120 to 180 lb.

Well, I got the necessary fittings and that afternoon made the change in the pipes. I also had to change the brush-holders around as the direction of rotation was reversed.

When all was ready to start up, a number of the "oil-field mechanics" came around to see the engine refuse to start, and to have a good laugh on me, but it started off quietly and ran perfectly cool. With a wide-open throttle the voltage did not vary 1 per cent., even with large variations in the load, and the exhaust could hardly be heard when standing at the end of the exhaust pipe.

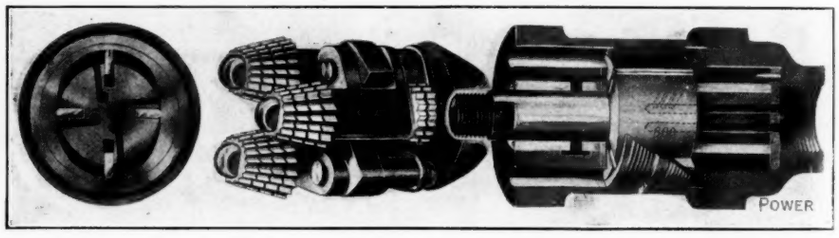
While I had the steam-chest cover off, I took the valve out and made a diagram of it and measured the parts. Then I put the valve back and turned the engine over by hand in the direction it had been running, and noticed how the different events took place. Steam had been admitted for full stroke, cutoff occurring just as the crank passed the center; it was then compressed about ½ stroke, when the release took place and it had to exhaust in the remaining half stroke. I did not have the good luck to own an indicator at that time and I shall always regret it, as a diagram from that engine would have been a souvenir worth framing.

On the second night I made some more discoveries. The boilers also supplied steam for one well and two large pumps for pumping the oil to the railroad, 10 miles away. These were started whenever the tanks were full, by an attendant who had to look after twelve or fifteen pumps scattered all over the surrounding oil fields. No warning was given when he was ready to start a pump, but when it had made about four strokes, I knew it, as the engine slowed down and stopped. I noticed before it stopped that there was a steady stream of water from the exhaust. I tried to keep the engine running by carrying just one gage of water in the boilers, but it would pull over and eventually stop the engine.

The next day I got a piece of 6-in. pipe threaded at both ends and about 5 in. long; also, two 6-in. caps, one with two holes drilled and tapped for a 1½-in. pipe, and the other for one 1½-in. pipe. These caps were put on the 6-in. pipe and the latter connected in the steam line to the engine, as shown in Fig. 1. The drain valve was left cracked, but the water came over so fast that the engine slowed down no matter what was done. Therefore, I decided to change the connection at the top of the boilers. In Fig. 2 are shown both the original and altered connections, the former at *A* and the latter at *B*. After the change was made no more trouble was experienced from wet steam at the engine.

## New Lagonda Air Driven Boiler Tube Cleaner

The sectional cuts of this cleaner, shown herewith, give a good idea as to its construction. The compressed air or steam passes through two ports in a plate in the rear end of the cleaner, then through transverse openings in the rotor, and out through branch openings to the space behind the paddles. There are but two ports opening into the air chamber and only the two paddles doing the work are under air pressure.




LAGONDA AIR DRIVEN BOILER TUBE CLEANER

To secure economical operation under various air pressures and in different hardnesses or thickness of scale, two interchangeable rear plates are provided having different-sized port areas. Where a limited amount of air is available, the plate with the smaller holes can be used, securing plenty of power with a small amount of air. If the scale is heavy and plenty of air can be furnished, the plate with larger holes can then be used and more power will be developed.

The cleaner is manufactured by the Lagonda Manufacturing Co., Springfield, Ohio.

It is announced from Toronto that the government of Saskatchewan has made a contract by which the Canadian Pacific, Canadian Northern, and Grand Trunk railways will spend $20,000,000 diverting the South Saskatchewan River to supply water to Regina, Moose Jaw and other cities, and also in developing 30,000 electrical horsepower. A commission has been appointed to prepare plans.

# Development of Large Steam Turbines

**Discussion before the American Society of Mechanical Engineers of the paper by Prof. A. C. Christie on the above subject.**

**An abstract of this paper appeared in the July 23 issue, page 144, and an extract "Economy Tests of Steam Turbines," June 11, page 831.**

In the discussion of the paper upon the present state of development of large steam turbines presented by Prof. A. E. Christie, at the recent meeting of the American Society of Mechanical Engineers, Carl George DeLaval, general manager of Henry R. Worthington, called attention to the new radial-flow reaction turbine designed by the Messrs. Ljungstrom, of Stockholm, a description of which will be published in an early issue. The steam enters between two disks, passing from center to circumference between concentric blading rings. The two disks revolve with their shafts at the same rate of speed, but in opposite directions. At the end of each shaft an electric generator is attached. With this arrangement the relative speed of each blade becomes double that in the ordinary reaction turbine of the same rotative speed and diameter with the result that the number of blade rows becomes only one-quarter as great. The general result is a turbine of small dimensions for the power and several have been and are being built. The first was a 500-kw. turbine running at 3000 r.p.m. which gave an efficiency of 71.8 per cent. and a steam consumption of 8.75 lb. per b.hp.-hr. with steam at 175 lb. absolute and 250 deg. F. with a vacuum of 28½ in. Sizes from 100 to 7500 kw. with a speed of 3000 r.p.m. have been designed on this double-rotating principle. Single-rotating turbines of the same design have also been brought out, but with slightly less efficiency and about 10 per cent. increased steam consumption.

The highest recorded efficiency is 71.8 on an Erste Brünner turbine with steam at 156.2 lb., 1500 r.p.m., 15.82 lb. of steam per kw.-hr. The Ljungstrom 1000-kw. turbine shows 74.7 per cent. efficiency, taking the steam in front of the throttle valve, and 76.9 per cent. behind the throttle valve, with steam at 162 lb., speed 3000 r.p.m., pounds of steam per kilowatt-hour 11.55, or 8 lb. per b.hp. In starting the double-rotating turbine the two sides may not run at the same speed, but when a certain speed is reached and the exciting currents of the generators are strong enough, they will automatically synchronize and run as one unit. In case of accident one generator can be left idle and the other continued in use with, of course, a loss of efficiency.

CLARENCE P. CHRISSEY, of the General Motor Co., Detroit, suggested the following classification:

(*A*) Impulse turbines with only one velocity stage in each pressure stage.

(*B*) Impulse turbines with two or more velocity stages in each pressure stage.

(*C*) Impulse turbines in which all pressure stages do not have an equal number of velocity stages.

Classification of impulse turbines might be carried further to illustrate a point of construction often overlooked; i.e., the method of conducting the steam leaving the last buckets to the succeeding nozzles. Various constructions are:

(*a*) Impulse turbines in which the steam leaving the last moving bucket passes directly from the bucket into the succeeding nozzles without great loss of velocity.

(*b*) Impulse turbines in which the steam leaving the last moving bucket passes into the casing and after losing practically all of its velocity enters the next set of nozzles.

(*c*) Impulse turbines in which the steam leaving the last moving bucket is utilized in some pressure stages, according to the methods described in (*b*), and in other pressure stages, according to the method outlined in (*a*).

It follows from this classification that

Rateau turbines fall in class *Ab*.

Zoelly turbines fall in class *Ac*.

Curtis turbines of General Electric Co. fall in class *Bb*.

Curtis turbines of A. E. G. (large size) fall in class *Cc*.

Professor Christie says that the single-row stages of the A. E. G. turbines are of the Rateau construction. The Allgemeine Electricitäts-Geselleschaft, however, states that these stages are constructed in accordance with the 1896 patent description of Curtis, which represents the immédiate forerunner of the Rateau and Zoelly construction. The difference between the A. E. G. single-row stages and those of Rateau consists in the manner in which the steam leaving the moving bucket is utilized, i.e., the A. E. G. single bucket row stages are of type *a* in which the steam leaving the moving bucket passes directly into the succeeding nozzle without great loss in velocity, while the Rateau falls under type *b* in which the steam is brought to rest before entering the next set of nozzles. The large A. E. G. turbine, therefore, appears to be of the Curtis type throughout, although not altogether of the type popularly known as Curtis.

The moving buckets of reaction turbines in practice have been supported upon drums and those of impulse turbines usually upon disks. The manner of construction does not, however, change the fundamental type, and therefore the author's designation of the Beliss and Morcom turbine as a Curtis-Parsons construction is confusing because this turbine operates entirely upon the impulse principle.

The paper conveys the impression that high steam velocities are inherent to impulse turbines having two or more velocity stages. Such is not necessarily the case; if low bucket velocities are employed the spouting velocity of the steam is also low. Marine turbines of the impulse type illustrate this point.

The theoretically correct action of the steam in any impulse turbine appears only at one set of conditions. For any variation in conditions, or quantity of steam flowing, a variation from the theoretically correct action must occur. The flat economy curves of impulse turbines prove that this is not important from the standpoint of efficiency, and practice shows that the end thrust is not appreciably increased by departure from the theoretical conditions.

The drum construction in impulse turbines was probably first used by the A. E. G. in the turbine for the steamship "Keiser," which was tested in 1905. The firm's experiments led them to the conclusion that the drum construction was advantageous for the low bucket speeds existing in direct-connected marine turbines when the steam had expanded to fifty times its initial volume.

While the efficiency ratio comes nearer than any other single feature to expressing the economic results, it must be used with caution because

1. The steam conditions must be taken into account.

The same steam turbine operating with an equal amount of available energy will not have the same efficiency ratio if the energy is obtained in one case by a moderate vacuum and high superheat, and in another case by high vacuum and low superheat.

A noncondensing turbine may have the same or a better efficiency ratio than a condensing machine, but this is no indication that the noncondensing turbine is as suitable or better for a given plant than a condensing unit.

2. It is more difficult to obtain a high efficiency ratio with a large amount of available energy than with a smaller amount.

Referring to the table of tests,* it is seen that the 6000-kw. machine built by Erste Brünner, M. F. G. has an efficiency ratio of 71.3 and that the heat available per pound of steam is

*Power, June 11, p. 832.

380.7 B.t.u. The A. E. G. turbine of about the same rating gives an efficiency ratio of 68.7, but the heat units available per pound of steam are 434.2 or about 14 per cent. greater. If the latter machine were operated with steam having 380.7 B.t.u. per lb., there is little doubt that the efficiency ratio would equal or approach closely that of the Erste Brünner turbine.

3. The efficiency of the electrical generators is included in the efficiency ratio.

This has considerable importance if the ratios are used to compare turbine types.

It is stated in the paper that European builders generally guarantee better generator efficiency than American builders.

4. The results may be affected by the inclusion or exclusion of the power necessary for steam excitation and auxiliaries, as well as other variations in testing practice.

Speed is a factor which cannot be neglected in making comparisons upon which to base opinions as to type. Either type is benefited by an increase of rotative speed, and it is almost certain that in the future the speed of all types will not differ materially for machines of a given rating and service.

E. D. DREYFUS, of the West Penn Railways Co., urged the establishment of some method of turbine rating, which would be recognized by, and satisfactory to, the engineering fraternity, also rules of conducting tests of this type of power-generating unit, including the turbine and generator as a whole. There is a wide dissimilarity in present practice.

PROF. JAMES A. MOYER, of the University of Michigan, said that such a tabulation as Professor Christie had presented would be incomplete if it did not include the very elaborately detailed and accurate test of a Sulzer turbine of the combined impulse-reaction type reported by Probessor Stodola in the *Zeitschrift des Vereines deutscher Ingenieure,* Oct. 28, 1911, page 1799. At the load showing the best economy (2058 kw.) the steam consumption was 15.21 lb. per kw.-hr. Speed was 1501 r.p.m., absolute pressure 178.15 lb., superheat 182 deg. F., vacuum, referred to 29.92 in. barometer, 28.41 in. (0.740 lb. per sq.in. abs. pressure). Calculated from these data the B.t.u. per kw.-hr. are 18,800, the heat utilized per pound of steam 224.5 B.t.u., the available heat per pound of steam 380.1 B.t.u. and the efficiency ratio 59.1 per cent. The generator efficiency was low for a machine of this kind. This is certainly a reliable test of a presumably well designed turbine of the Curtis-Parsons type and the results are not particularly favorable as compared with the data ordinarily published. It should be clear, therefore, that the adoption of a design consisting of both impulse and reaction stages along with excellence of construction details, does not necessarily give higher efficiencies on the Rankine cycle than other types. The Sulzer turbine was laboring under the disadvantage that it had three rows of moving blades and two rows of stationary blades instead of the modern Curtis construction, employing two rows of moving and one row of stationary blades per stage. It has never been shown that anything is gained by attaching to a blade wheel more than two rows of moving blades or buckets.

Professor Moyer felt that tests showing Rankine efficiencies greater than about 68 per cent. should be accepted only with a great deal of caution. He did not feel justified in accepting without reservation the data of the first three and the fifth tests reported in the table. The second of these tests does not bear the earmarks of a test made with very much accuracy. Practically all the data are given in round numbers, usually indicating approximation. The load is exactly 6000 kw. The steam pressure is given roughly at 13 atmospheres absolute. The steam temperature averaged apparently exactly 300 deg. C. at the throttle throughout the test, and the steam consumption per kw.-hr. was calculated not much closer than the nearest half pound. He was inclined to agree with Mr. Emmett in his statement at a previous meeting of the society that the data for the fourth test were "not altogether convincing." There is very little erosion of the blades in either the Curtis or the so-called Curtis-Parsons type, and the constant erosion of continuous operation for five years will have very little effect on the economy of the turbine as compared with the corresponding deterioration of a reciprocating engine. He criticized Mr. Zoelly's claim that he got the best results on his turbine by throttle governing. The Zoelly method of governing is cheap and simple and that, as he understood it, is the reason for its adoption. He would like to be shown, however, by authentic tests that it is even as good as some other types.

GEORGE A. ORROK, of the New York Edison Co., stated that he had been sufficiently interested in the test which Professor Moyer questioned to correspond with the maker of the test and had thoroughly satisfied himself as to its authenticity.

FRANCIS HODGKINSON, of the Westinghouse Machine Co., said that for 60-cycle work from 3000 to 10,000 kw., 1800 r.p.m., is the speed employed by the Westinghouse company. For larger powers than this, 1200 r.p.m. is used. For 25-cycle work 1500 r.p.m. is employed up to a rating of 15,000 kw. normal, or 20,000 kw. maximum. A speed of 750 r.p.m. is not likely to be again employed for capacities less than 30,000 to 40,000 kw. The impulse blading is operated at speeds of from 350 to 500 ft. per sec., according to the particular type or size of machine. The reaction or Parsons blading occasionally reaches a speed of 500 ft. per sec. at the extreme low pressure end in cases of high powers at the given speeds. The higher speed being necessary because of the large steam volumes involved, the last low-pressure rows varying in condensing machines from 300 to 500 ft. per sec. In non-condensing machines, of course, the velocities are lower.

Considerably higher journal velocities are employed than Professor Christie has shown; 80 ft. per sec. and 100 lb. per sq.in. of projected area are commonly employed. There can be no question that such pressures and velocities may be materially exceeded without encountering wear of the bearings, if proper bearing design, and flooded lubrication are employed. The higher the journal velocity, the greater is the rate of shear of the oil, and hence more cooling capacity must be employed. However, it is undesirable to operate with cold oil, as this only increases the bearing losses. No difficulties are encountered with temperatures approximating 150 deg. F.

Professor Christie quotes the opinion of European engineers that higher economy is to be obtained from impulse elements of the Rateau or Zoelly types, than with impulse elements of the Curtis design. Mr. Hodgkinson thought this to be a generally accepted fact, the greater losses in the case of the latter type being doubtlessly due to the low efficiency of the last or third row of moving blades, probably because of eddies produced by the steam passing through the previous rows. A previous speaker had stated that the third moving row in a Curtis element had never been found of value. This is not true for certain high steam and low blade velocities. It is certain, however, that the efficiency of each succeeding moving row rapidly decreases.

By the same token, it would seem that the reaction turbine may be more economical than either, unless it be subjected to some disability. This would seem to be plain, for with an impulse turbine two complete conversions of energy are necessary—the first expanding the steam through nozzles, the steam doing work in giving itself velocity, the second energy transformation being in the blades, where the velocity of the steam is abstracted. Both of these transformations of energy are subject to losses, the second greater than the first. If the nozzles could be permitted to revolve, and there were no blades at all, obviously such a turbine would be the more efficient, if, as said before, it does not suffer from some disability or disadvantage. The reaction or Parsons turbine is a machine of this type, but unfortunately is subject to some leakage. However, where the speed of the turbine permits a de-

sign of blading which has an inappreciable ratio of leakage, such a turbine is highly efficient, which is exactly the feature of some of the later high-speed turbines, and is also the feature of the combination type of turbine, comprising a Curtis element for the high-pressure, and a reaction element for the low-pressure section. The Curtis element, as compared to an all Parsons design, replaces a large number of rows of reaction blading of small diameter, with the attendant great length and weakness of structure, or a smaller number of rows of larger diameter in which case the blades will be smaller and of poor proportions, subject to high rates of leakage.

Of late, the tendency has been toward higher speeds, which, within reasonable limits, is conducive to better results, not only from the standpoint of economy, but also from the standpoint of operativeness. As Professor Christie has stated, congestion at the low-pressure end may be entirely avoided by making the last portion double-flow. Higher rotational speed involves smaller diameters and hence better blade proportions at the high-pressure end with the attendant lesser leakage. In addition to this, the turbine cylinder structure as a whole is much less gigantic, which tends toward less distortion of structure with temperature changes, thus permitting, if it were desired to do so, running the blades with smaller clearances. Higher rotational speed does not necessarily mean higher stresses in the rotating parts.

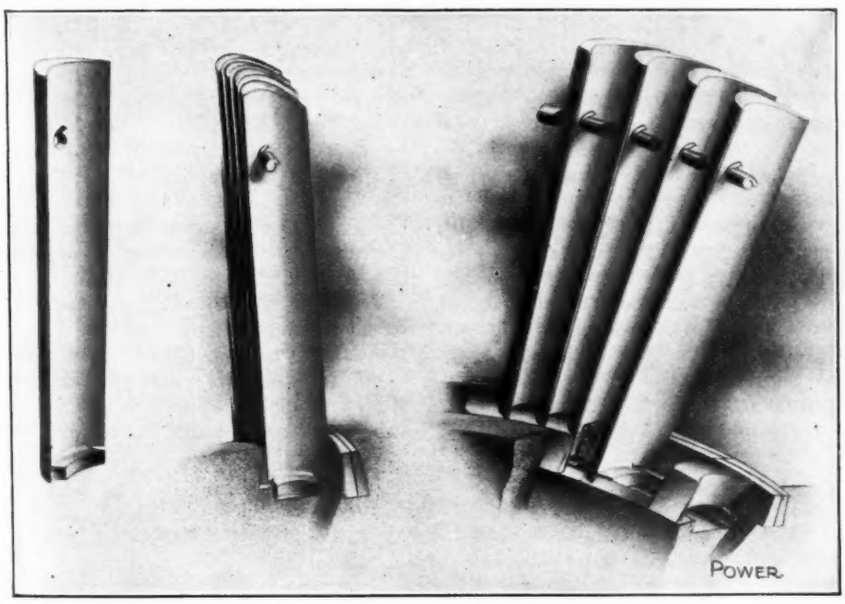
Professor Christie very properly points out that there is a lower leakage ratio in a single-flow turbine, than in one of double-flow construction. However, the choice of rotative speed is limited on account of the standard frequencies employed. While reducing the speed to the next lower one available may permit of single-flow construction, the advantages may be entirely offset by the increased clearances which must be employed, simply because the size of the machine has so materially increased with the attendant greater amount of distortion. Therefore, the turbine having the low-pressure sections double-flow is frequently more economical. It is convenient to be able to double up the steam passage at the low-pressure end, because of the tremendous rate with which the steam volume increases when turbines are designed for high vacuum.

Concerning blade troubles and blade construction, Mr. Hodgkinson ventured the statement that no turbine built by a reputable manufacturer in this country, ever came to grief because of the blades becoming detached, due to centrifugal force, per se. Accidents have happened because of misalignment or distortion which have resulted either in blades colliding with turbine parts or with one another, which of necessity resulted in injury. The principal source of blade trouble is due to the blades buzzing or vibrating in the steam current. The natural period of the blade may be raised by means of a shroud or other form of lashing, which, of course, offers some protection. One thing to be avoided in blade construction is the reduction of section where the blade is attached to the blade-carrying member, which obviously permits of deflection due to vibration taking place at that point, which if material, results in ultimate fracture.

Mr. Hodgkinson exhibited with the aid of the lantern slide reproduced herewith, standard construction for reaction blading as employed by the Westinghouse Machine Co. The feature, as will be seen, is an absolute interlocking system, the strength of which is equal to that of the blade itself. The blade sections are stronger at the root than at the tip. The section tapers slightly from the root, thus eliminating chance of injury due to vibrating or buzzing, should it exist.

As shown in the illustration, the bottom end of the blade is upset, by a truly upsetting process and not any mere bending, and the die for holding the blade at this time, is so arranged that the section of the blade is increased in thickness throughout its lower portion, this latter feature, however, not being clearly shown in the illustration.

The dovetailed groove in the blade-carrying member has a supplementary groove in the floor of the main groove into which fits the upset end of the blade. The latter locks under the packing pieces which occupy the main groove. The larger blades have compound wedges which go between the walls of the groove and the blades and packing pieces, effectively filling the groove more satisfactorily than a calking strip. With the smaller blades, the wedges are found unnecessary, owing to the lighter sections of the packing pieces, which permits them to fill the groove, when driven up into place. Inasmuch as no calking of any kind is resorted to, blades may be taken out and replaced an indefinite number of times.




FORM OF BLADING USED IN WESTINGHOUSE TURBINE

Professor Christie discusses overload capacities of machines as practiced in this country and Europe, and suggests that the society fix a rating for turbines, defining their overload capacity. This may be difficult unless the turbine rating be entirely disassociated from that of the generator. The speaker cordially concurred with Professor Christie in that much confusion exists. In discussing capacities of machines, one is obliged to go into some explanation as to what one means. In fact, the rating of machines needs more describing nowadays than does the load factor. It causes particular confusion in statements of the cost, per kilowatt, of installations.

There are essentially two kinds of ratings required today, in modern power plants.

One case is that of the large plant containing many units, where a unit if running at all, is carrying its maximum load. The unit best adapted for such a plant is one the generator of which has a maximum continuous rating with say a 50-deg. rise, the turbine to give its best steam consumption at this load. It need have no overload capacity, or at most but 10 or 15 per cent.

The other case is that of smaller plants, containing fewer units, and having to encounter certain daily peaks, when it is convenient to have units capable of carry-

ing loads for limited periods at some sacrifice of economy. Here, while the generator may still be given a certain maximum continuous rating, the turbine should be most economical at some fraction of this load. The old system of rating a machine at a normal full load at 40 deg. rise, the turbine giving best economy at that load, the generator being capable of carrying 25 per cent. overload continuously, and both being capable of carrying 50 per cent. overload for a limited time, is a system of rating, admirably adapted for the plant containing few units. This system of rating was used in this country in the early days of turbine work, and had not been previously used in Europe. The system was brought about by the turbine having to compete with the Corliss engine of that day. Latterly, "bargain-counter methods," as Professor Christie calls them, have caused such units to have their turbines sometimes given a maximum continuous rating corresponding to the 50 per cent. overload just referred to. It would seem entirely possible to have uniformity of generator ratings on a maximum continuous load basis—the turbine, however, to have its best steam consumption at some fraction of this rating, and overload capacity to suit the generator. Plainly, the generator and turbine should be considered separately.

W. L. R. EMMET, of the General Electric Co., criticized the turbine tests as tabulated by Professor Christie, which he claimed do great injustice to the Curtis type and convey very incorrect impressions.

The first machine mentioned in this table is a 2000-kw. Curtis-Parsons machine operating at 1500 r.p.m. and is reported to give an efficiency of 71.9 per cent. with a moderate degree of superheat. Everybody familiar with the design of either Curtis or Parsons machines knows perfectly well that no such result has ever been produced under such conditions.

The efficiencies assigned to the five General Electric Curtis machines mentioned range from 63.6 down to 61 per cent. The first, which operates at 3464 kw., is a representative result of the performances of the machine in question and is a good performance when the very high vacuum used is considered. Of the other four only one of the tests is representative of the performances of the machine in question when correctly tested and in good condition, and in that case allgrees of vacuum than it is with very high degrees of vacuum, the limitations to the effective use of high vacuum being many and their extent being affected by speed and capacity. Therefore the designer who produces a relatively low efficiency with a very high vacuum may have accomplished his purpose much more creditably than another who has produced a higher efficiency with a low vacuum.

None of the General Electric machines referred to in Professor Christie's tabulation, except the first mentioned, was designed later than 1904, while many of the other makes are of recent production. Mr. Emmett presented a tabulation of tests (Table I) of a few representative Curtis machines. The first two tests in this list apply to machines in Boston, very carefully and repeatedly tested when they were comparatively new. These are of the same date and type as most of those to which the figures in Professor Christie's tabulated report could apply, and are representative of the General Electric's original large five-stage vertical units when correctly tested and in good condition. The tests were made by the Boston Edison Co.'s engineers with very perfect facilities.

The third test applies to an 8000-kw. machine of the same date and type in Chicago. One of the tests in Professor Christie's paper also purports to refer to this machine under different conditions.

The next three items given refer to recent six-stage Curtis machines and the last to a three-stage Curtis machine recently built by the British Thomson-Houston Co. This machine produces a very remarkable result, considering its small capacity and extreme simplicity.

TABLE I. TESTS OF CURTIS TURBINES

| Customer | Date of Test | Load K.W. | R.p.m. | Steam Press. Abs. | Superheat Dg.F. | Vacuum | Water rate lb. per kw. hr. | Efficiency per cent. |
|---|---|---|---|---|---|---|---|---|
| Boston | | 7526 | 720 | 184.40 | 134 | 28.65 | 13.732 | 66.6 |
| Edison Co. | 1907 | 7481 | 720 | 185.10 | 129 | 28.58 | 13.866 | 66.6 |
| Commonwealth Edison Co., Chicago | 1907 | 12000 | 750 | 200.00 | 125 | 28.00 | 14.220 | 66.2 |
| Boston Edison Co. | 1912 | 12460 | 720 | 207.50 | 191 | 27.52 | 13.630 | 68.2 |
| Stock | 1911 | 4000 | 1800 | 189.00 | 0 | 28.00 | 16.050 | 64.3 |
| Louisville Ltg. Co. | 1911 | 6500 | 1800 | 180.00 | 100 | 28.00 | 14.780 | 66.10 |
| British Thomson-Houston (maker) Rugby | 1912 | 2000 | 3000 | 165.00 | 150 | 28.00 | 14.400 | 67.3 |

TABLE II. ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES
EFFICIENCY RATIOS BASED ON E.H.P. MARKS & DAVIS STEAM TABLES USED
References ÷ Zeit. D.V.D. Ing.—Zeitschrift Des Vereines Deutscher Ingenieure

| Maker of Turbine | Type | Date of Test | Load K.w. | R.p.m. | Steam Pressure Lbs. Absolute | Temperature at Throttle–deg.F. | Vacuum Referred to 29.92" Bar | Condenser Pressure Lb. Absolute | Lbs. of Steam per Kw.-hr. | B.t.u. per Kw.-hr. | Heat Utilized per lb. of Steam | Heat Available per lb. of Steam | Efficiency Ratio | Reference |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Escher, Wyss & Co. | Zoelly | 1912 | 1,275 | 3000 | 196.3 | 588 | 28.99 | 0.455 | 12.59 | 15,940 | 268.6 | 418.1 | 64.3 | Zeit. D.V.D. Ing. 4/13/12 |
| Escher, Wyss & Co. | Zoelly | 1912 | 5,418 | 1500 | 180.7 | 588 | 28.74 | 0.580 | 11.95 | 15,060 | 285.5 | 402.1 | 71.0 | Zeit.D.V.D. Ing. 4/13/12 |
| Elsassische Maschinenbau Gesellschaft | Zoelly | 1912 | 10,006 | 1250 | 167.8 | 565 | 28.65 | 0.625 | 12.61 | 14,975 | 284.2 | 389.1 | 73.0 | Zeit. D.V.D. Ing. 4/13/13 |
| Belliss & Morcom | Drum Impulse | 1911 | 3,000 | 1500 | 192.5 | 558 | 27.92 | 0.981 | 15.0 | 18,420 | 227.3 | 372.7 | 61.0 | Manufacturer's Data |
| Belliss & Morcom | Drum Impulse | 1911 | 2,500 | 1500 | 164.7 | 466 | 26.92 | 1.471 | 16.95 | 19,815 | 201.2 | 323.6 | 62.2 | Manufacturer's Data |
| Aktiebolaget Ljungstrom Angturbin | Radial Reaction (Ljungstrom) | 1911 | 1,000 | 3000 | 181.7 | 669 | 28.55 | 0.671 | 11.68 | 15,130 | 292.0 | 414.9 | 70.4 | Engineering 4/19/12 |
| | | 1911 | 972 | 3000 | 176.7 | 667 | 28.58 | 0.657 | 11.55 | 14,960 | 295.3 | 414.2 | 71.3 | |

In discussing this table the writer says, "It is therefore apparent that the efficiency ratio alone will best express the degree to which the designer has approached ideal results in his turbines." To this statement exception should be taken for the reason that it is naturally possible to produce higher efficiencies in turbine construction with moderate degrees of vacuum than it is with very high degrees of vacuum, the limitations to the effective use of high vacuum being many and their extent being affected by speed and capacity. Therefore the designer who produces a relatively low efficiency with a very high vacuum may have accomplished his purpose much more creditably than another who has produced a higher efficiency with a low vacuum.

The accurate testing of steam units requires great care and is susceptible of many kinds of error. Many figures concerning such tests, which are entirely incorrect, are being constantly circulated. It is only by comparing many tests and carefully observing their consistency as to results and characteristics that one can ever be sure as to turbine economy. The figures given in the accompanying table have been so analyzed and com-

pared as to be unquestionably correct, and it will be observed that they give an impression concerning the value of Curtis turbines which is very different from that produced by Professor Christie's tabulation.

C. V. KERR, chief engineer, McEwen Bros., said with regard to wear of the blading due to wetness of the steam, that in his double-bucket turbine they had found the wear of the blading in the upper stages. That may be due to one or both of two things. In the first place they allowed a greater proportion of the available energy to produce velocity in the upper stages with the idea that part of that would be transferred to the lower stages and thus equalize the wear further down. This may have been overdone.

Another reason is that the steam is denser in the upper stages and that its erosive power is greater. With regard to the rating of steam turbines, there is one point which is a sort of corollary of the rating question. The small steam-turbine builders drop into the habit of selling their machines on the power basis. A turbine is a peculiar machine. The casing can be made for a certain manufacturing cost and by adding nozzles the power can be doubled, or trebled, or quadrupled, but the small steam turbine competes with the steam engine and the electric motor and they are generally sold on the power basis; hence it is the practice now with at least two of the small steam-turbine builders, to establish a minimum price for the turbine, based on manufacturing costs and then rate upward on the basis of additional power development.

In his closure the author added the following tests as supplementary to the table printed with his paper:

## At a Salt Works Power Plant

BY E. J. SAXE

I was employed as an expert engineer by a large salt company which operated a number of widely scattered plants. My job was to get the largest amount of salt from a given amount of coal.

The works to which I first went contained 11 grainers, or shallow wooden tanks, each 150 ft. long, 14 ft. wide, and 18 in. deep, lined with tile, and containing 1500 ft. of 4-in. pipe, laid lengthwise and connected by return bends; they were suspended from timbers laid crosswise of the tanks so that they were about 4 in. under the surface of the brine when the tanks were full.

The boiler room contained twelve 72-ft. by 18-in. horizontal return-tubular boilers, each having a 60-ft. steel stack; they were hand fired, using slack coal for fuel. In an adjoining room there were the necessary pumps for feeding the boilers and pumping the brine; nearby were two ancient engines. The plant was making about 7 bbl. of salt per ton of coal, but the first expected 10 bbl.; the possibilities were 13 bbl.

For the first week I stood by and watched the process and soon discovered that the only way to make salt was the way it was being made; any change would mean: "You can't make no salt."

The draft gage showed 0.4 in. at the base of the stack, and from 0.25 in. to about nothing in the firebox. The Orsat apparatus showed 3 to 4 per cent. $CO_2$; the stack temperature averaged 650 deg. F.

Metering the water showed that the boilers were doing 200 hp. when the firemen were feeling good, and I found that one grainer would condense all the steam one boiler could make.

The firemen received $1.80 per day of 12 hours, and the plant ran 24 hours per day, 365 days per year. I got a mason to stop up the cracks in the boiler setting, and put in arches and wing walls under the boilers, and got a better showing in the draft readings in the furnace and a much higher $CO_2$. I tried to stir up the firemen to do better work. But not much enthusiasm could be obtained on a 12-hour shift paying $1.80 per day.

Matters had begun to look up and as less coal was being burned and more steam made I turned my attention to the grainer room. The grainers were set higher than the boilers, and an 8-in. pipe ran the length of the room and supplied the 4-in. pipe with steam. A 1¼-in. pipe drained the condensation into a 4-in. return pipe and thence into a return tank, from which it was taken by the feed pump and fed to the boilers. The condensation flow was regulated by hand, and to make sure that all the water was drained from the coils, steam was blown through them and discharged to the atmosphere through the vapor pipe on the return tank.

I proposed the use of traps, but was informed that, "If you do we can't make no salt." However, I got the traps, and then ran against a problem. The brine was kept at a temperature of about 198 deg. F. When the coils were clean, 10 lb. of steam would give the desired temperature, but as the coils accumulate scale, the pressure was increased until, when the scale became ½ in. thick, 100 lb. pressure was required to do the work. This condition would occur in from two to three weeks, when the brine was drawn off and the scale removed. The problem was to get a trap that would handle the water with this variable pressure, but a trap with a large balanced valve was selected, and did the work satisfactorily. Steam was saved and salt was still made.

My next discovery was that while the return water came to the return tank at 212 deg. F., the feed water was only heated about 120 deg. An investigation showed that a 2-in. pipe from the bottom of the tank was wide open to the sewer, and cold water was making up the supply; I was told that the feed pump would not handle the water any hotter. The pump should have handled the water as the steam tank was 18 in. higher than the pump, and there was some reason if it would not. Investigation disclosed that the water end had been designed for valves ½ in. thick, but someone had substituted ⅝-in. valves, with extra-heavy springs, and the valves had a scant ⅛-in. lift. New valves were secured, light springs used, and the pump did pump hot water. The result of all these small economies was the desired 13 bbl. of salt per ton of coal.

The fuel used was a very volatile, long flaming coal, and the smoke that poured from the stacks was a constant eyesore, but with various arrangements of brick arches and wing walls, I could not stop it without using steam jets, which I would not do. I could not persuade the firm to pull down the stacks, and put in induced draft, which would have eliminated the smoke, and made a saving of at least 10 per cent. of the fuel.

While making the changes I ran boiler tests to determine the advantage of each. My first test was plain furnace versus arch and wing walls; result, 13 per cent. economy and 13.5 per cent. increased capacity in favor of the wing walls and arches.

Next I picked out a fireman who was willing to do as he was told, and had him fire the boiler as carefully as possible; result, 27 per cent. increase in capacity and 10 per cent. in economy.

Then I told the manager that he was burning 90 tons of coal per day, at a cost of $225. I said: "If I can give those firemen $1 per day to stand up to the boilers and fire them with three shovelfuls at two-minute intervals, they will be glad to do it. It will save 10 per cent. or a profit of $22.50 at a cost of $8; net $14.50 per day. Do away with the smoke, and have the firemen looking for the job instead of the job always looking for the firemen." But he said: "It can't be done."

With such decisions on the part of those higher up, the engineer does not find much encouragement or opportunity for introducing economical methods of operation.

In 1910 the mines of the Transvaal employed 257 steam-driven air compressors having 100,291 hp., there being an average of three 50-drill compressors on each mine. Formerly, all the air compressors were steam-driven, but now electric energy is furnished to all the mines by the Victoria Falls & Transvaal Power Co., Ltd., and electricity for air compressors has superseded steam.

# Electrical Department

**Conducted to be of service to the men in charge of electrical equipment in the power house**

## Inspecting and Testing Electrical Apparatus—II

By A. L. Cook

### Alternating-current Systems

Alternating-current generating systems generally may be divided into three classes:

1. Generation at 220, 440, or 550 volts for supplying power and light to small areas or to a single building. The system may be three-phase, quarter or two-phase, or single-phase; the frequency generally 60 cycles.

2. Generation at 1100 or 2200 volts for supplying power and light a distance of several miles from the station. As in class 1 the system may be single or polyphase, generally with a frequency of 60 cycles.

3. Generation at 6600, 11,000 or 13,200 volts for supplying power and light to distances not exceeding 15 miles. The three-phase system is generally used for power systems, for this class, at a frequency of 60 or 25 cycles.

To transmit power more than 15 miles, still higher voltage is necessary for economy, and step-up transformers are required, since generators are not wound for voltages higher than those of class 3. All alternating-current generators require direct-current exciters and these should be inspected and tested as described in the previous article for direct-current machines.

### Low Voltages

In class 1, the switchboard is often similar to that for direct-current, laminated-brush, carbon-break circuit-breakers and standard knife switches being employed, with the busbars mounted at the rear of the board. The present tendency is to use oil circuit-breakers with the higher voltages, such as 440 or 550, particularly if the station is of large capacity. The switchboard should be inspected like that for direct current, as already described, either a magneto or a direct-current voltmeter being used to test for grounds.

Oil circuit-breakers should be examined for good contact when the switch is closed, and automatic circuit-breakers should be tripped several times by raising the plunger by hand. Circuit-breakers which are tripped by current transformers should preferably be tested with current passed through the trip coil. Usually these coils require from 5 to 10 amp. to trip the switch.

Sometimes the switches are closed and tripped by direct current, in which case the current transformers would be connected to a relay which closes the tripping circuit. Usually all the trip coils or relays are wound for the same current. The current in the feeder, causing the circuit-breaker to open, depends upon the ratio of the current transformers connected in the feeder circuit. Thus, if the transformers have a ratio of 1500 to 5 amp., 5 amp. would flow in the trip-coil circuit when 1500 amp. is flowing in the feeder. Usually an ammeter is in series with the trip coil so that it receives 5 amp. at the same time, although the scale generally indicates the actual current in the feeder. Usually the trip coil is marked for the actual current in the feeder the same as the ammeter.

If a source of alternating current of the same frequency as that normally used on the feeder, is available, the trip coil may be tested by passing current through the secondary circuit. Fig. 3 shows connections for current transformers and trip coils as used in two-phase and three-phase circuits. To make the test, the secondaries of the current transformers should be first short-circuited, as indicated by the dotted lines; then the circuit should be opened at a convenient point, such as *A*, and the test circuit inserted. The trip coil, feeder ammeter and test circuit must all be in series. The alternating-current supply may be at 110 volts, or any other convenient voltage, and the amount of current required usually will not be more than 10 amp. The resistance in the test circuit may consist of incandescent lamps or a small water rheostat. The trip-coil plunger should be set at the desired point and the current gradually increased with the rheostat and the feeder-ammeter reading noted when the circuit-breaker opens. This is the current which must flow in the feeder to trip the circuit-breaker.

### Medium Voltages

For voltages of 1100 or 2200, oil circuit-breakers are nearly always used. The busbars are frequently at the back of the switchboard, although it is better to mount them and circuit-breakers away from the board, with only the operating handles on the panels. The circuit-breakers should be inspected as outlined under class 1, and if opened and closed electrically, should be operated several times to make sure that they close and trip satisfactorily and that the proper indicating lamp lights each time. The relays used with these breakers may be tested as described for ordinary circuit-breaker trip coils.

Tests for grounds on this class of equipment cannot be made very satisfactorily with a magneto or an ordinary voltmeter, but, if nothing better is available, either may be used to indicate the condition of the insulation. If possible,

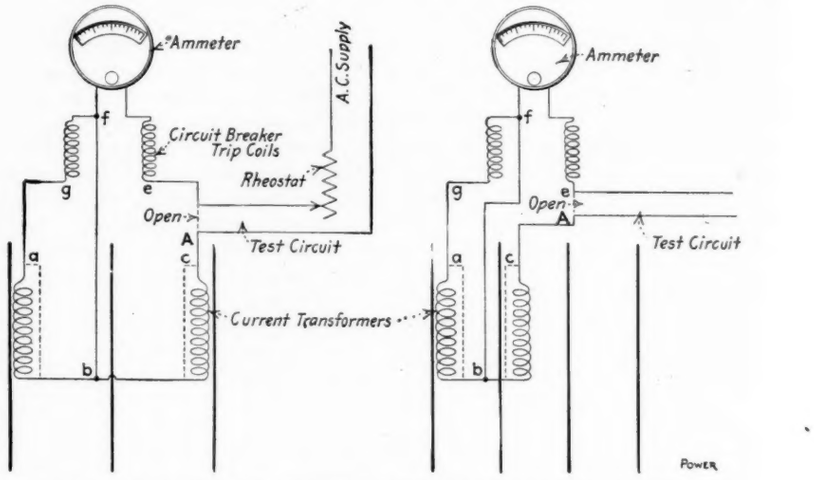



Fig. 3. Connections for Current Transformers and Trip Coils for Three- and Two-phase Circuits

the wiring should be tested with high voltage. For important systems a test with double the normal voltage, applied for 15 min. between each conductor and the ground is satisfactory. This test voltage is to be applied only to the wiring and not to the generators or the voltage transformers. The oil circuit-breakers and switches, however, can be tested with the wiring. Since the current transformers are connected in with the wiring, they would also be subjected to this test voltage. The current transformers usually furnished for this service are insulated to withstand a test of this character, but if there is any doubt about it the manufacturer of the transformers should be consulted before making the test.

Voltage tests generally can be made only on new systems, for extensions to stations in operation as usually arranged make this impracticable. If the new part of the equipment can be isolated from the old, tests should be made. If no apparatus is available, giving the required voltage for testing the new equipment, as described, a simple test can be made with one of the main generators. The wiring to be tested should be connected to a terminal of one of the generators which is disconnected from the busbars. A second terminal of the generator is then grounded. As a precaution against a rush of current if the insulator breaks down it is sometimes advisable to ground this terminal through resistance. The generator voltage is raised slowly until the voltmeter connected to the two generator terminals reads about 10 per cent. above normal at which point it should be held for about 15 min. This also subjects the generator to a voltage slightly greater than normal, but this voltage is not dangerous for machines of standard construction.

### High Voltages

In the third class as in the second, oil circuit-breakers are usually employed with single-pole knife switches in addition for isolating the circuit-breakers and other apparatus for inspection and cleaning. The busbars and circuit-breakers are seldom mounted on the switchboard, but are installed remote from the board, usually with the individual circuit-breakers and the busbars inclosed in masonry compartments. The inspection of the circuit-breakers should be made as outlined for the medium-voltage system.

For high-tension apparatus, it is desirable to test the station wiring for faults, which can be done satisfactorily only by subjecting the equipment to a high-voltage test. This requires a testing transformer unless regular transformers of the proper voltage are available. A test at double the normal voltage applied for 15 min. between the wiring and the ground is sufficient. In such tests, as previously mentioned, the wiring, busbars, switches, oil circuit-breakers and current transformers, constitute the apparatus to be tested; the other equipment, such as generators and transformers, should not be subjected to this test since they are either tested in the factory or, if wound at the plant, would be given special tests by the erectors.

To make a voltage test on the wiring, a relatively low voltage is first applied, such as 10 per cent. of the final value, and gradually increased until the proper value is reached. If a regular testing outfit is used, generally some method is provided for adjusting this voltage, but if an ordinary transformer is used special provision must be made for controlling the voltage. Most convenient is to use a generator supplying only the testing transformer, regulating the generator voltage by its field rheostat. Another method is to employ a water rheostat in series with the primary of the testing transformer, but this regulation is more difficult and less satisfactory for testing purposes.

The actual voltage applied to the apparatus may be determined; by reading the voltage on the low-tension side of the transformer with a voltmeter and multiplying by the ratio of the transformer as marked on the name-plate; or by measuring the actual testing voltage with an electro-static voltmeter or a spark gap. The first method gives only approximate results but is sufficiently accurate for ordinary tests, especially if the testing load is not more than one-fourth the full-load rating of the transformer. The second method, although more accurate, requires more careful manipulation. If a high-voltage test can be made as described, the wiring at least should be tested by using the generator alone as described for medium voltages.

The bearings should be examined and the air gap checked as explained for direct-current machines. The brushes used on the slip rings should be fitted and the spring tension adjusted. The machine should then be freed from dirt, using air under pressure, if available.

### Testing the Generator

If possible, the generator insulation should be tested by the voltmeter method before being put in service. With the machines belonging to class 1 in apparently good condition, this test is not very important, but for the other two classes is essential. Voltmeters which operate either on direct or alternating current are not suitable for making insulation tests, the only proper kind being the permanent-magnet type which can be used only on direct current. The insulation resistance required for alternating-current armatures is given by one manufacturer as approximately 1,000,000 ohms per 1000 volts of rated voltage, when the machine is at its full-load temperature. The field winding, however, should show a resistance higher than this. Large machines will have a lower resistance than small machines of the same voltage rating. For machines in the first and second classes, the ordinary 125-volt circuit and the 150-volt instrument can be used to test the insulation, but for the third class a high-resistance voltmeter is generally required. This class of machine is frequently given a high-voltage test, but this should be made only under the direction of the manufacturer.

If the generator has been exposed to moisture, it must be dried out and some provision made for testing the insulation as outlined before. To dry out the generator, the armature should be short-circuited beyond the main switch and current transformers, and the field current adjusted to give sufficient armature current so that the temperature shall not exceed 160 deg. F. on any exposed part of the windings. Anyway, the current should not exceed 150 per cent. of full-load current. If sufficient temperature rise is difficult to get at normal speed, owing to the small iron loss, the speed may be reduced, thereby decreasing the air circulation through the ventilating ducts; or with inclosed turbo-generators, the duct may be partly closed. High-voltage machines take longer to dry out than low-voltage machines, due to the greater amount of insulation. Arrangements should be made to measure the insulation resistance from time to time as described for direct-current machines. (Article I.)

Before starting a polyphase generator the main connections should be checked, if possible, for proper phase relation, so they will not have to be changed when the generator is "phased out." With two machines of the same make, corresponding terminals should connect to the same busbar, if the generators rotate in the same direction when viewed from the collector-ring side. If they rotate in opposite directions, any two of the leads of a three-phase machine, or the two leads of one phase of a two-phase machine, should be interchanged.

Before starting the generator, the main switch and field switch should be open and the field rheostat all cut in. The generator can then be started slowly. The oil rings should be inspected and the generator then can be gradually brought to speed. The field switch should be closed and the voltage raised to normal by cutting out the field rheostat. In a three-phase generator the voltages of each phase should be the same. With a two-phase four-wire machine, the voltage of the two phases should be the same. For a two-phase, three-wire generator, the voltage between either outside terminal and the common terminal

should be the same, and that between outside wires should be 1.41 times this value.

If the station has only one generator, the load can now be thrown on, but if one must operate in parallel with other

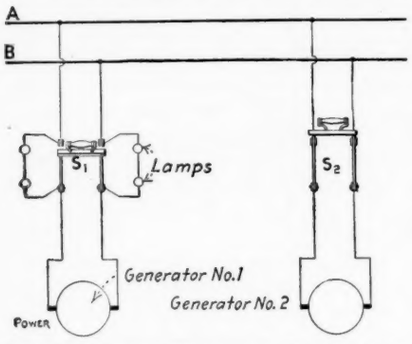



FIG. 4. SHOWING METHOD OF SYNCHRONIZING LOW-VOLTAGE GENERATORS

machines, it is necessary to "phase out" the new machine and also check the synchonizing connections.

### PUTTING GENERATORS IN PARALLEL

For two alternating-current generators to operate properly in parallel, the voltages must be approximately equal, the frequencies the same, and the voltages in phase, that is, the corresponding maximum and minimum values of the voltage must occur at the same instant; under these conditions the machines are in synchronism.

The fundamental operation of synchronizing is illustrated for a single-phase generator by Fig. 4. Generator No. 2 has its main switch $S_2$ closed and is therefore connected to the busbars $A$ and $B$; generator No. 1 has switch $S_1$ open and is to be synchronized with the busbars. Incandescent lamps are connected across the terminals of the switch, as shown, the number in series on each side being such that the sum of their voltages equals the voltage of the generator. When generator No. 1 is in synchronism the lamps will be dark, since there will be no difference in potential across the corresponding terminals of the switch. If the voltage of generator No. 1 is approximately the same as that of the busbars, but the frequency is not the same, the lamps will vary at intervals from bright to dark and the rapidity of these pulsations will be an indication of the difference in frequency between the busbars and generator No. 1. The speed of the generator can then be changed until the pulsations of the lamps occur quite slowly, and when this condition is reached and the lamps are dark, generator switch $S_1$ can be closed and the generator is thus put in parallel with the busbars.

For low-voltage machines, such as are included in class 1, lamps could be used in this way, but for higher voltages, such as those of classes 2 and 3, potential transformers must be used. Fig. 5 is a diagram of connections for this case. It will be seen that the conditions are similar to those in Fig. 4. Usually the transformers have such a ratio that normal voltage on the generators gives about 110 volts on the secondary; therefore two 110-volt lamps are sufficient. If the frequency of generator No. 1 is not the same as that of the busbars, the lamps will pulsate, and when the lamps are dark for some time the generator is in phase and switch $S_1$ can be closed. If the secondary connections are crossed, the lamps will burn brightly when the generator is in phase, instead of being dark as before; this arrangement is preferred by some operators.

For polyphase machines, the conditions outlined should occur on each phase simultaneously. Thus for a three-phase machine, if three sets of lamps are connected across corresponding terminals of the

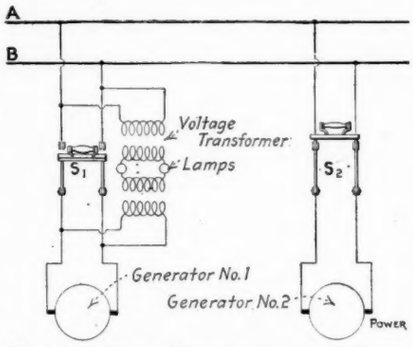



FIG. 5. SHOWING METHOD OF SYNCHRONIZING HIGH-VOLTAGE GENERATORS

main switch, all three sets of lamps should be dark and bright together. Having once phased out a polyphase machine, the generator thereafter need only be synchronized on one phase, unless there is some way by which the main connections of the generator or the busbars can become changed.

it to another machine which is known to have correct connections and to bring the two machines up to speed together. If the station can be shut down for a time, the two generators may both be connected to the busbars by closing the main switches; if not, a spare generator can be used with temporary connections. These temporary connections should be made so that when the switches are closed, the machines are paralleled the same as if connected to the busbars. The fields of each should now be excited, adjusting the field current to about that required to give normal voltage at no load when the generator is up to speed. With the two armatures thus connected the generators should be started slowly, keeping both as nearly as possible at the same speed. At first heavy current will flow in the machines, but as they speed up, this should decrease and finally fall to a small amount indicating synchronism. This should occur before half speed is reached. If the heavy current continues and seems to increase, the connections are wrong and the two leads of one phase, in the case of a two-phase machine, or any two leads of a three-phase machine, should be interchanged, and the test repeated. The generators should fall into synchronism as indicated by the ammeters and the machines can be brought up to normal speed together.

If the machines are both connected to the busbars in the foregoing test, the permanent synchronizing connections can be tested at the same time. When the main connections have been found correct, the synchronizing lamps or synchroscope should indicate that the machines are in synchronism, and if not, the synchronizing connections should be changed. If, however, temporary connections were made in phasing out, the synchronizing connections must be checked separately. To do this, the main

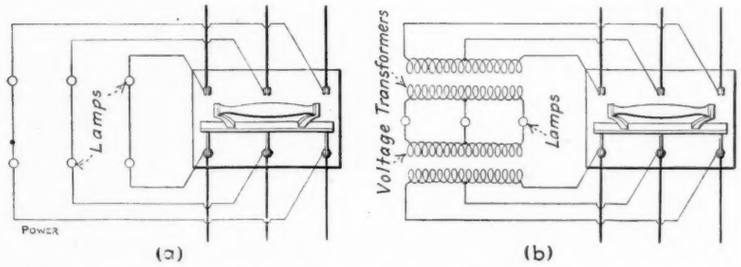



FIG. 6. SYNCHRONIZING CONNECTIONS FOR THREE-PHASE GENERATORS

At the time that the machine is phased out, the synchronizing connections may be checked. With single-phase generators, if the synchronizing connections can be easily traced, no further test is necessary, since there are only two main leads and they can be synchronized as previously described. For polyphase generators, especially where potential transformers are used, the easiest way to phase out the new machine is to connect leads of the new generator should be disconnected at the terminals, and the ends of the leads fastened clear of each other, and the main generator switch closed, thus connecting the leads to the busbar. This will also connect the synchronizing circuits of the machine, and the lamps should indicate synchronism. If they do not, the synchronizing connections should be changed. Care should be taken to see that these synchronizing

connections are made to the corresponding phase of the busbars and machine.

When it is not convenient to bring two machines up to speed together to phase out, temporary synchronizing connections may be made. This method requires connections for each phase similar to those shown in Figs. 4 or 5. For three-phase generators the proper connections are shown in Fig. 6, *a* being for low-voltage machines and *b* for high voltage. The connections are made across the main switch of the generator to be tested. If the generator is brought up to speed and its voltage adjusted to normal, the three sets of lamps should be bright and dark simultaneously, the rapidity of changes from dark to bright depending upon the difference in the frequency of the machine and busbar. If the lamps are not

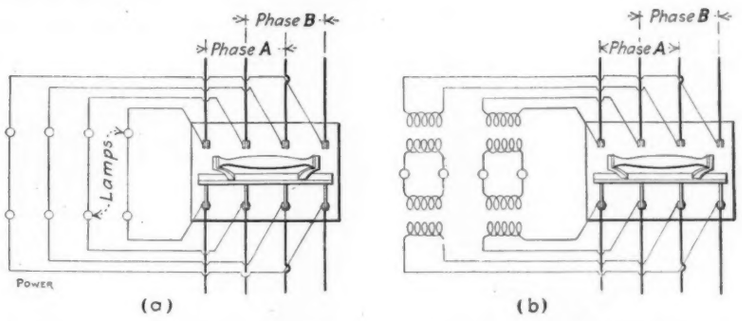



FIG. 7. SYNCHRONIZING CONNECTIONS FOR A TWO-PHASE, FOUR-WIRE GENERATOR

all bright simultaneously, any two of the main leads should be interchanged, keeping the synchronizing connections on the same terminals of the switch as before, and the test repeated. The lamps should now all fluctuate together, indicating that the main connections are correct. Fig. 7 shows similar connections for a two-phase, four-wire generator. The leads corresponding to each phase are first determined by a voltmeter as previously described. When the generator is operated, the lamps should all burn brightly at the same time; if not, the two leads of either phase should be reversed.

A polyphase induction motor may be used to phase out a generator, in which case the motor is first connected to the terminals of the main switch on the busbar side, through transformers if necessary, and the direction of rotation noted. The motor leads are marked and then connected to the generator side of the switch, taking care that the leads in each case are connected to the same pole. If the motor rotates in the same direction the connections are correct. Having phased out the main generator connections by either of these methods, the permanent synchronizing connections must be checked by disconnecting the generator leads at the terminals of the machine, and closing the main switch as previously described.

Each time that the machine is put in service during regular operation the generator must be synchronized before it is paralleled with the other machines. To do this only requires adjusting the speed of the incoming machine until the regular synchronizing devices indicate that it is in proper phase relation with the busbars, and then the main switch can be closed.

Incandescent lamps are frequently used to indicate synchronism, as described, but many stations have synchronism indicators, or synchroscopes, which have a pointer mounted in a suitable case, with a dial marked at one point indicating synchronism. Usually in these instruments, the pointer is free to rotate and has no springs to control its action. When one pair of terminals of this instrument is connected to the incoming generator and the other pair to the busbars, the pointer will rotate if the two generators do not have the same frequency, and the speed of rotation will depend upon the difference in frequency. If the incoming machine is slightly slower than the other generators, the pointer will rotate in one direction, and if faster, will rotate in the opposite direction, clearly indicating what adjustment of the machine is necessary. The instrument is usually connected in the secondary circuit of voltage transformers, one pair of terminals being connected to each transformer. The lamps in Fig. 5 can be retained as a check, even if the synchroscope is used.

If the generator is in multiple with other generators, the load can be transferred to the new machine by increasing the speed of the latter slightly. A change in the field current would have practically no effect upon the amount of load that the generator would carry, but would change the current in the generator armature. In shutting down an alternating-current generator, it is preferable to transfer as much of the load as possible to the other machines, but frequently this is not done; the main switch being opened with the generator carrying its share of the load. The field switch should *never* be opened while the generator is under load and connected in parallel with other generators, as a very heavy flow of current would result.

## Chicago Electrolysis Ordinance

The city council of Chicago passed an ordinance July 15 requiring the railways to so equip and operate their systems that damage due to electrolysis caused by stray currents will be prevented.

The ordinance requires that all uninsulated electrical return circuits must be of such current-carrying capacity, and so arranged, that the difference of potential between any two points on the return will not exceed 12 volts, and between any two points on the return 1000 ft. apart, within one-mile radius of the City Hall, will not exceed 1 volt, and between any two points on the return 700 ft. apart, outside of this one-mile radius limit, will not exceed 1 volt. In addition, a proper return conductor system must be so installed and maintained as to protect all metallic work from electrolysis damage.

The return-current amperage on pipes and cable sheaths must not be greater than 0.5 amp. per lb.-ft. for calked cast-iron pipe; 8 amp. per lb.-ft. for screwed wrought-iron pipe, and 16 amp. per lb.-ft. for standard lead or lead-alloy sheaths of cables.

The railways must equip their uninsulated return-current systems with insulated pilot wire circuits and voltmeters so that accurate chart records will be obtained daily showing the difference of potential between the negative busbars in each station and at least four extreme limits on the return circuit in its corresponding feeding district.

The system must also be equipped with recording ammeters, insulated cables and automatic reverse load and overload circuit-breakers which will record and limit the maximum amperes drained from all the metallic work (except the regular return feeders) to less than 10 per cent. of the total output of the station.

The chart records must be so kept as to be always accessible to city officials.

Failure to comply with the ordinance within 9 months from its passage will incur a fine not less than $50 nor more than $200 for each offense; and each day's operation of such equipment contrary to the provisions shall constitute and be regarded as a separate and distinct offense.

The ordinance is to take effect at once.

Ray Palmer, electrician for the city, after extensive observation and numerous careful tests, has estimated that the damage to the pipe lines of the city and the gas company and to the cable sheaths of the city, sanitary district of Chicago, and the telephone company due to stray currents from the tunnel, surface and elevated railway systems, represent an annual maintenance cost of upwards of $600,000. In addition vast damage is done to the steel and iron work of office buildings and other structures which cannot be estimated.

# Gas Power Department

**Worth-while gas-engine and producer information treated in a way that can be of practical use**

## American Practice in Rating Internal Combustion Engines—IV*

By T. C. Ulbricht and C. E. Torrance, Jr.

### Inlet and Exhaust Valves

In this article equations will be given for the relations between valve diameters, cylinder diameters, valve lifts, etc., as derived from the accompanying curves.

Curve 2, Fig. 29, represents the average values of inlet valve diameter plotted against cylinder diameter, for horizontal engines up to 30 in. cylinder diameter. Above this size the ratio of valve to cylinder diameters increases more rapidly and approaches curve 1, which represents maximum values for the smaller engines.

This is because in sizes above 30 in. the ratio of stroke to cylinder diameter increases. This is especially true from 28 or 30 in. to 48 in. diameter, where the piston speeds are high; since the diameter of the inlet valve is plotted against the cylinder diameter, which does not increase as fast as the piston speed, the inlet-valve diameter must increase more rapidly in proportion to the cylinder diameter, to supply a greater quantity of fuel. Letting $d$ equal the diameter of the cylinder and $d_v$ the diameter of the valve, the curves in Fig. 29 may be represented by the formulas:

$d_v = 0.45\ d$ (maximum) curve 1;
$d_v = 0.345\ d$ (average) curve 2;
$d_v = 0.25\ d$ (minimum) curve 3.

Fig. 30 shows similar curves for the exhaust valves of horizontal engines. The equations are:

$d_v = 0.40\ d$ (maximum) curve 1;
$d_v = 0.35\ d$ (average) curve 2;
$d_v = 0.29\ d$ (minimum) curve 3.

Fig. 31 shows average curves for the inlet and exhaust valves of vertical engines, the equation $d_v = 0.34\ d$ being the same for both inlet and exhaust valves.

Average curves plotted between valve lifts (inlet and exhaust) and corresponding valve diameters, for horizontal and vertical engines, are shown in Figs. 32 and 33.

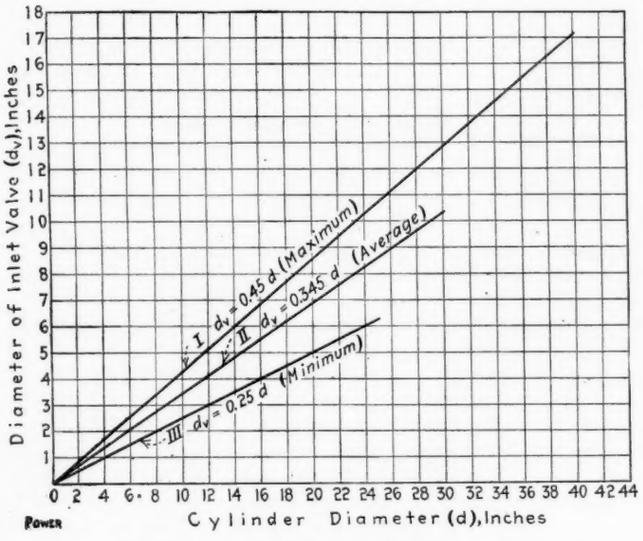

Diameter of Inlet Valve ($d_v$), Inches
Cylinder Diameter (d), Inches
I $d_v = 0.45\ d$ (Maximum)
II $d_v = 0.345\ d$ (Average)
III $d_v = 0.25\ d$ (Minimum)
POWER

Fig. 29. Inlet Valve Diameter vs. Cylinder Diameter for Horizontal Engines

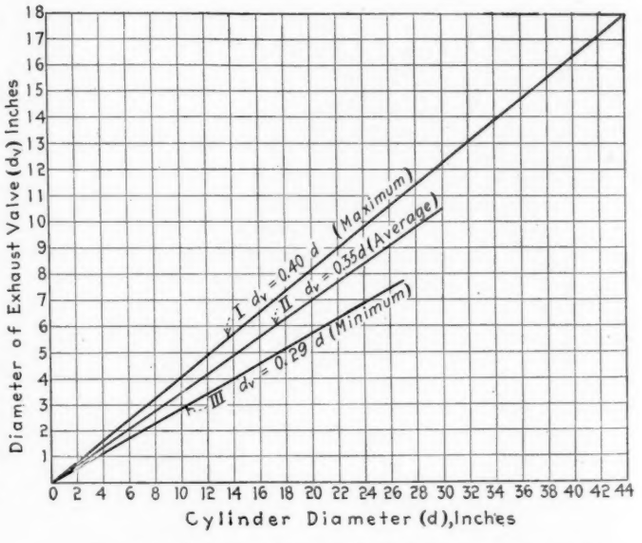



Fig. 30. Exhaust Valve Diameter vs. Cylinder Diameter for Horizontal Engines

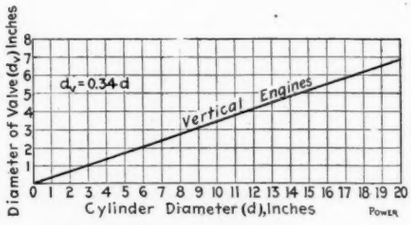



Fig. 31. Cylinder Diameter vs. Valve Diameter for Both Inlet and Exhaust Valves of Vertical Engines

### Gas Velocities through Inlet and Exhaust Ports

When a quantity of gas $Q$ passes through an opening of area $F$ with a velocity $V$,

$$Q = FV$$

where $Q$ is in cubic feet per minute, $F$ in square feet, and $V$ in feet per minute.

The total charge taken into and exhausted from the engine per minute equals the quantity of gas passing through both the inlet and exhaust ports per minute; that is,

$$Q = \frac{\pi d^2}{4 \times 144} \times \frac{2\ ln}{12} = \frac{3.14\ d^2 \times l \times n}{2 \times 1728} = \frac{d^2\ ln}{1100}$$

where $d$ is the diameter of the cylinder in inches, $l$ the length of stroke in inches, and $n$, the revolutions per minute.

The area of the port opening in square inches is $F = \pi\ d_v\ l_v$, in which $d_v$ is the valve diameter in inches, and $l_v$ the valve lift in inches. Substituting these values,

$$F = \frac{3.1416\ d_v\ l_v}{144} = \frac{d_v\ l_v}{45.83}\ sq.ft.$$

but as previously found,

$$Q = FV = \frac{d^2\ ln}{1100}$$

hence,

$$\frac{d^2\ ln}{1100} = \frac{d_v\ l_v}{45.83} \times V$$

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and,

$$V = \frac{d^2 ln \times 45.83}{1100 \times d_v l_v} = \frac{d^2 ln}{24\, d_v l_v} \text{ ft. per min.}$$

Table 3 contains the velocities calculated by this method for various assumed values:

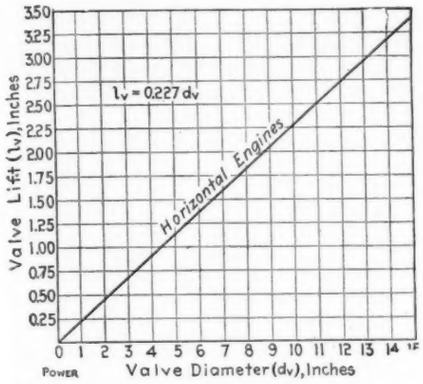



FIG. 32. VALVE LIFT VS. VALVE DIAMETER FOR BOTH INLET AND EXHAUST VALVES IN HORIZONTAL ENGINES

In each case for the assumed brake horsepower rating per cylinder per end, the corresponding $d^2ln$ value was found from the average $d^2ln$ versus *b.hp.* curves of article I. The revolutions per minute were then read from the *r.p.m.* versus *b.hp.* curves. Also, $d^2ln \div n = d^2l$ from which $d$ and $l$ can be found as follows:

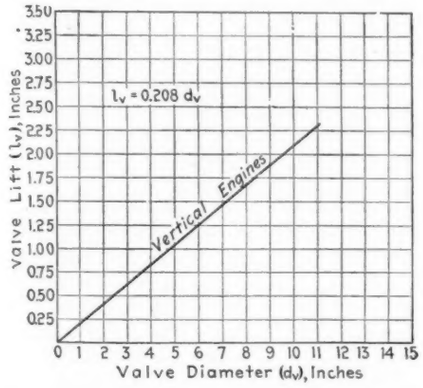
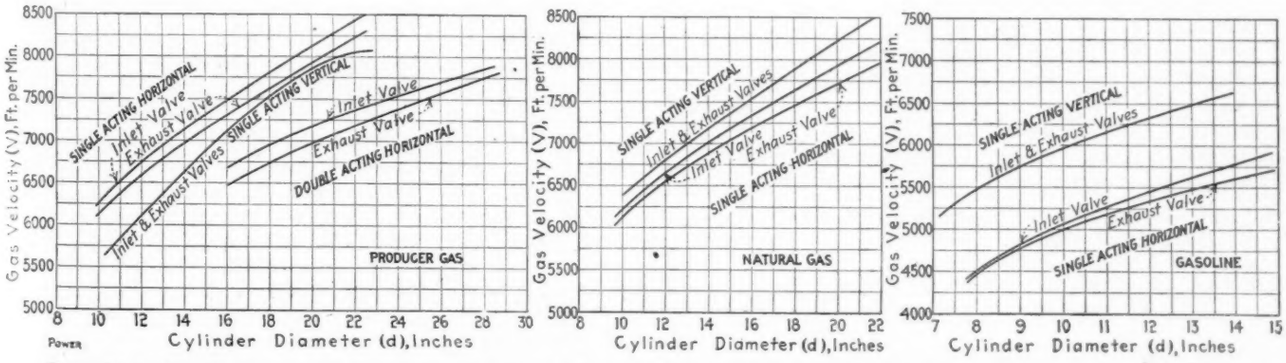



FIG. 33. VALVE LIFT VS. VALVE DIAMETER FOR INLET AND EXHAUST VALVES IN VERTICAL ENGINES

TABLE 3. GAS VELOCITIES THROUGH VALVES

| Rated b.hp. | R.p.m. | $d^2ln$ | $d^2l$ | $d$ | $l$ | $d_v$ Inlet | $d_v$ Exhaust | $l_v$ Inlet | $l_v$ Exhaust | Gas Velocity Inlet | Gas Velocity Exhaust |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **SINGLE-ACTING HORIZONTAL (GASOLINE)** | | | | | | | | | | | |
| 10 | 270 | 170,000 | 630 | 7¾″ | 10¼″ | 2.68 | 2.68 | 0.6 | 0.6 | 4400 | 4400 |
| 20 | 223 | 333,000 | 1,492 | 10¾″ | 14⅛″ | 3.5 | 3.5 | 0.78 | 0.79 | 5080 | 5020 |
| 30 | 200 | 500,000 | 2,500 | 12″ | 17⅛″ | 4.12 | 4.16 | 0.93 | 0.94 | 5440 | 5330 |
| 40 | 185 | 660,000 | 3,570 | 13¾″ | 19⅛″ | 4.7 | 4.76 | 1.07 | 1.08 | 5470 | 5350 |
| 50 | 183 | 825,000 | 4,500 | 14½″ | 21⅛″ | 4.98 | 5.05 | 1.12 | 1.15 | 6160 | 5920 |
| Avg.... | | | | | | | | Ft. per min. | | 5310 | 5204 |
| **SINGLE-ACTING HORIZONTAL (PRODUCER GAS)** | | | | | | | | | | | |
| 20 | 290 | 380,000 | 1,310 | 9¾″ | 13⅛″ | 3.35 | 3.38 | 0.76 | 0.77 | 6220 | 6080 |
| 50 | 225 | 920,000 | 4,080 | 14¼″ | 20⅛″ | 4.87 | 4.90 | 1.10 | 1.12 | 7150 | 6980 |
| 100 | 185 | 1,810,000 | 9,780 | 18⅛″ | 27⅛″ | 6.5 | 6.57 | 1.47 | 1.49 | 7900 | 7720 |
| 125 | 176 | 2,250,000 | 12,800 | 20⅛″ | 30″ | 7.10 | 7.20 | 1.61 | 1.63 | 8200 | 8000 |
| 150 | 171 | 2,700,000 | 15,780 | 22⅛″ | 32″ | 7.60 | 7.71 | 1.725 | 1.76 | 8580 | 8290 |
| Avg.... | | | | | | | | Ft. per min. | | 7610 | 7414 |
| **SINGLE-ACTING HORIZONTAL (NATURAL GAS)** | | | | | | | | | | | |
| 20 | 290 | 370,000 | 1,275 | 9⅛″ | 14″ | 3.32 | 3.33 | 0.76 | 0.77 | 6120 | 6010 |
| 50 | 225 | 830,000 | 3,690 | 13⅞″ | 20″ | 4.70 | 4.74 | 1.06 | 1.07 | 6940 | 6820 |
| 100 | 185 | 1,590,000 | 8,600 | 18″ | 26⅛″ | 6.20 | 6.25 | 1.42 | 1.43 | 7530 | 7420 |
| 125 | 176 | 2,040,000 | 11,580 | 20″ | 29″ | 6.90 | 6.98 | 1.55 | 1.58 | 7960 | 7710 |
| 150 | 171 | 2,330,000 | 13,620 | 21″ | 30⅛″ | 7.22 | 7.32 | 1.65 | 1.68 | 8140 | 7900 |
| Avg.... | | | | | | | | Ft. per min. | | 7338 | 7172 |
| **DOUBLE-ACTING HORIZONTAL (PRODUCER GAS)** | | | | | | | | | | | |
| 50 | 192 | 1,100,000 | 5,730 | 16″ | 22⅛″ | 5.5 | 5.58 | 1.25 | 1.27 | 6670 | 6470 |
| 100 | 146 | 2,120,000 | 14,500 | 21⅛″ | 32″ | 7.3 | 7.4 | 1.66 | 1.68 | 7300 | 7100 |
| 125 | 134 | 2,640,000 | 19,700 | 23⅛″ | 36″ | 8.1 | 8.2 | 1.83 | 1.87 | 7420 | 7170 |
| 150 | 125 | 3,160,000 | 25,250 | 25⅛″ | 39⅛″ | 8.7 | 8.8 | 1.98 | 2.00 | 7650 | 7480 |
| 200 | 113 | 4,200,000 | 37,100 | 28⅛″ | 45⅛″ | 9.84 | 9.95 | 2.23 | 2.25 | 7900 | 7820 |
| Avg.... | | | | | | | | Ft. per min. | | 7388 | 7208 |
| **SINGLE-ACTING VERTICAL (GASOLINE)** | | | | | | | | | | | |
| 10 | 397 | 170,000 | 428 | 7⅛″ | 8⅛″ | 2.46 | 2.46 | 0.56 | 0.56 | 5140 | 5140 |
| 20 | 332 | 333,000 | 1,000 | 9⅛″ | 11⅛″ | 3.25 | 3.25 | 0.73 | 0.73 | 5850 | 5850 |
| 30 | 296 | 500,000 | 1,690 | 11⅛″ | 13″ | 3.90 | 3.90 | 0.876 | 0.876 | 6110 | 6110 |
| 40 | 275 | 660,000 | 2,400 | 12⅛″ | 14⅛″ | 4.36 | 4.36 | 0.98 | 0.98 | 6440 | 6440 |
| 50 | 260 | 825,000 | 3,170 | 14″ | 16″ | 4.80 | 4.80 | 1.08 | 1.08 | 6620 | 6620 |
| Avg.... | | | | | | | | Ft. per min. | | 6032 | 6032 |
| **SINGLE-ACTING VERTICAL (PRODUCER GAS)** | | | | | | | | | | | |
| 20 | 332 | 380,000 | 1,215 | 10⅛″ | 11⅛″ | 3.52 | 3.52 | 0.80 | 0.80 | 5630 | 5630 |
| 50 | 260 | 920,000 | 3,540 | 14⅛″ | 16⅛″ | 5.02 | 5.02 | 1.13 | 1.13 | 6760 | 6760 |
| 100 | 223 | 1,810,000 | 8,120 | 19⅛″ | 21⅛″ | 6.65 | 6.65 | 1.48 | 1.48 | 7780 | 7780 |
| 125 | 215 | 2,250,000 | 10,450 | 21″ | 23⅛″ | 7.21 | 7.21 | 1.63 | 1.63 | 7980 | 7980 |
| 150 | 208 | 2,700,000 | 13,000 | 22¼″ | 25⅛″ | 7.82 | 7.82 | 1.78 | 1.78 | 8080 | 8080 |
| Avg.... | | | | | | | | Ft. per min. | | 7246 | 7246 |
| **SINGLE-ACTING VERTICAL (NATURAL GAS)** | | | | | | | | | | | |
| 20 | 332 | 370,000 | 1,113 | 10″ | 11¼″ | 3.41 | 3.41 | 0.71 | 0.71 | 6380 | 6380 |
| 50 | 260 | 830,000 | 3,190 | 14¼″ | 15.7″ | 4.80 | 4.80 | 1.00 | 1.00 | 7200 | 7200 |
| 100 | 223 | 1,590,000 | 7,130 | 18⅛″ | 20.8″ | 6.35 | 6.35 | 1.33 | 1.33 | 7840 | 7840 |
| 125 | 215 | 2,040,000 | 9,480 | 20⅛″ | 23″ | 7.00 | 7.00 | 1.47 | 1.47 | 8275 | 8275 |
| 150 | 208 | 2,330,000 | 10,600 | 21⅛″ | 23¼″ | 7.30 | 7.30 | 1.53 | 1.53 | 8700 | 8700 |
| Avg.... | | | | | | | | Ft. per min. | | 7679 | 7679 |




FIGS. 34 TO 36. SHOWING THE RELATION BETWEEN CYLINDER DIAMETER AND GAS VELOCITIES FOR ENGINES RUNNING ON PRODUCER GAS, NATURAL GAS AND GASOLINE, RESPECTIVELY

For present purposes let the value of $d^2l = a$ or $l = \frac{a}{d^2}$. Then substitute this in the equation showing the relation between diameter and stroke for the particular type of engine and kind of gas as derived from Fig. 12, article I. For a single-acting horizontal engine using any fuel this is

$$d = 0.677l + 0.4$$

Substituting for $l$,

$$d = 0.667\frac{a}{d^2} + 0.4$$

or

$$d^3 = 0.667\ a + 0.4\ d^2$$
$$d^3 - 0.4\ d^2 = 0.667\ a$$

Assume that $a$ or $d^2l = 630$, then

$$d^3 - 0.4\ d^2 = 0.667 \times 630 = 420$$

whence

$$d = 7¾\ in.$$

and since $d^2l = 630$, $l = 10½$ in.

After finding $d$ and $l$, the corresponding inlet and exhaust valve diameters and lifts for the given cylinder diameters are read from the curves in Figs. 29 to 33, which show the relations between $d_v$ and $d$, and $d_v$ and $l_v$. These values are then substituted in the equation for $V$, namely,

$$V = \frac{d^2\,ln}{24\ d_v l_v}\ ft.\ per\ minute$$

The curves in Figs. 34, 35 and 36 show the foregoing calculated velocities in feet per minute plotted against cylinder diameter for the various gases.

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Attention is called to two typographical errors appearing in the first of the series of articles "American Practice in Rating Internal Combustion Engines" in the July 23 issue. The tenth line of the third column, page 121, should read

$$b.hp. = \frac{d^2\,ln}{16{,}400} - 0.5\ \text{(Curve 2, Fig. 5)}$$

Also on page 123, near the bottom of the third column, the diameter of the cylinder should be 19 in. instead of 16 in.

## Observations on Cooling Cylinders

By James H. Beattie

A good gas engine will convert into work from 25 to 30 per cent. of the heat in the fuel burned, 70 to 75 per cent. being radiated or lost in the exhaust. About 30 per cent. of the total heat of the fuel is radiated from the cylinder jacket and this must be carried away by some cooling medium, usually water, but sometimes air or certain grades of oil. Engines having jacketed cylinders for water-cooling are by far the most common, air-cooling being used only for small cylinders, while oil is not so well adapted as a cooling medium on account of its inability to take up and give off heat as rapidly as water.

Several types of water-cooled engines are in use; the most common is the ordinary closed jacket, in which the water is circulated either by a thermo-siphon system, or by a pump. When the pump is used, it is employed to assist the natural tendency of the heated water to rise in the system and allow the cold water to replace it. Among the arrangements for cooling the water after it is heated by passing through the engine cylinders, the oldest and most common, for stationary engines, is the circulating tank, but this has the disadvantage that it must be very large for a comparatively small engine. A 25-hp. engine requires a tank of at least 2000 gal. capacity. Other devices for cooling the water more rapidly include cooling sprays, radiators, such as used in automobile engines, and motor-driven fans, which force a strong blast of air through the spray.

An improperly cooled cylinder will wear rapidly and interfere seriously with the operation of the engine, for lubrication is made difficult and preignition, with pounding and excessive speed fluctuation, is likely to occur. The thermal efficiency of an engine is higher the hotter the cylinder; hence, for economy, the cylinder should be kept near the highest allowable temperature. For the ordinary closed-jacket engine of moderate size, the discharged water should be at 160 to 180 deg. F. With hopper-cooled, or open-jacket engines, the temperature is usually about 212 deg. F. The temperature of air-cooled engines is higher still, but all the engines designed to run at high temperatures have more allowance in the fit of the pistons and rings to permit greater expansion.

Accidents due to improper cooling are extremely common, especially with small engines, where the attention given them is limited. The following cases illustrate how lack of attention to these details often leads to serious results. A small horizontal portable engine had been in constant use for several weeks, and on the day the accident occurred had been in operation for 14 hr. As it was some distance from the machine it was driving, it received only an occasional visit from the operator. The stud bolts holding the cylinder to the frame sheared off, and the cylinder blew back, breaking the connecting rod and the piston. Examination showed that the jacket was nearly filled with lime, the oil cup was empty and the piston and cylinder perfectly dry. The piston had seized the walls of the cylinder so strongly that portions of the metal between the piston rings had been torn off. Fortunately the cylinder was uninjured and by fitting new stud bolts and a new piston and connecting rod, the engine was again in good condition. The lime was removed by placing a 10 per cent. solution of muriatic acid in the jacket and allowing it to stand over night. After this accident, rain water was substituted for the well water previously used, and more attention was given the oil cups.

The operator of another engine complained that it overheated badly. It had been in constant operation for several years and had always given excellent satisfaction. The cooling system consisted of a large tank and a belt-driven circulating pump, taking make-up water from a nearby stream. Investigation proved that the jacket was clogged with leaves and sediment drawn into the tank from the stream. The pump was able to keep up some circulation, but not sufficient to properly cool the engine.

In a similar instance, the trouble was from practically the same cause, but occurred in a different way. The engine was a small portable type, used for operating a sawmill, and overheated badly, burning out the cylinder-head packing. Leaves falling from nearby trees, it was found, had entered the jacket, clogging up the space so that water could not circulate, and causing the overheating. For those not familiar with the open-type jacket, it should be stated that it has an extension, or hopper, large enough to hold a gallon or two of water per horsepower. This water evaporates, the steam carrying away the heat, and the water is replaced from time to time as required. This system is used only on engines up to 25 or 30 hp., and is very economical and convenient, owing to the small amount of water necessary.

Another operator complained that the exhaust pipe of his engine and the exhaust-valve chest kept very hot, but the cooling system was found to be all right. The engine used excessive fuel for the work which it was doing, and the trouble proved to be in the carburetor. The needle valve regulating the flow of gasoline to the cylinder had been jammed so hard that the brass valve seat was burred until the opening to the air intake was larger than originally; consequently, the opening of the needle valve remaining the same, more fuel was allowed to pass than when the engine was new. An exceedingly rich mixture resulted and accounted for the large fuel consumption and also for the exhaust-valve casing and exhaust pipe overheating, as the mixture was still burning when the exhaust valve opened near the end of the power stroke. Substituting a new seat for the needle valve remedied the difficulty.

The adjustments of the valves, the carburetor and the ignition of the air-cooled engine are important. The owner of an air-cooled engine complained that it would run but a short time without overheating. His exhaust valve was found to open too late in the stroke, which was responsible for the overheating. The valve should have opened with the crank at least 30 deg. from the end of the power stroke, but was opening almost on center.

Other cases have been found where overheating was due to a combination of all three factors, improper valve timing, improper adjustment of the carburetor and too late ignition.

# Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

## Means for Accumulating Refrigeration—II

By Charles H. Herter

To date the effect of air agitation upon $k$, the coefficient of heat transmission per degree wall temperature difference, has received scant attention, and it is to be hoped that those having the knowledge and facilities to solve this interesting problem will soon attack it.

In April, 1909, Walter Kennedy, Pittsburgh, tested Nonpareil corkboard 2 in. thick, using a fan in and outside of the test box and obtained $k$ = 3.2 and 3.3 B.t.u. for the 2-in. thickness.

In 1911, John E. Starr tested 2-in. Union corkboard, using ice in the box and no fan, and derived $k$ ranging from 3.31 to 3.48 B.t.u.

In 1910, Prof. G. F. Gebhardt, Armour Institute of Technology, Chicago, tested Nonpareil and Union corkboard under identical conditions and found their conductivity $c$ to be exactly alike.

In 1910, Prof. F. L. Pryor, Stevens Institute of Technology, Hoboken, N. J., made tests on Star corkboard. A box built of 2-in. corkboard, with incandescent lamps inside, was placed in a room at 31 deg. F. Using the fan in the box, $k$ averaged 3.595 B.t.u.; and without the fan, 3.428 B.t.u. When the 2-in. box contained ice, and was placed in a room at 80 deg. F., $k$ without a fan averaged 3.495 B.t.u. That these tests are not conclusive as to the effect of air agitation on $k$, is shown by results from tests on a box of 4-in. Star corkboard which were conducted at the same time: With the box in the cold room, using heat and a fan in the box, $k$ was 1.74 B.t.u.; and without the fan, 1.736 and 1.772 B.t.u. through the 4-in. thickness.

From the above it would seem that the difference between $c$ and $k$ might be neglected, but the following additional data show that a difference exists that should not be neglected.

H. Recknagel's investigations on stone walls show $a_1$ to coincide approximately with the values obtained from hot-water and steam-heating pipes, or as in Table 2.

In analyzing other tests, where both the surface and the air temperature were actually taken and no effort was made to agitate the air, the writer found $a_2$ over a vertical plastered wall to increase with the insulating value of the wall, as follows:

| $c$ B.t.u. | $k$ B.t.u. | $a_2$ B.t.u. | $t_3 - t_4$ deg. F. |
|---|---|---|---|
| 5.68 | 3.904 | 24.95 | 7.1 |
| 3.99 | 3.102 | 27.97 | 5 |
| 1.939 | 1.739 | 33.6 | 2.42 |

Prof. Pryor's tests furnish a direct contradiction to this, his values of $a_1$ and $a_2$ (combined coefficients of radiation and convection at the warm and the cold side of wall respectively) with 4-in. cork figuring out to about half of the values found with 2-in. cork. In his test on 2-in. Star corkboard, where $k$ = 3.495 B.t.u., the rate of heat flow $H$ was

$$3.495 \times 26.4 \ deg. = 92.27 \ B.t.u.$$

and as the air was still, $a_1$ may be taken

TABLE 2. RECKNAGEL'S INVESTIGATIONS ON STONE WALLS

| Velocity of air at 1 ft. from wall | | $a_1$ in B.t.u., per sq.ft. per deg. diff., $t_1 - t_2$, per 24 hr. |
|---|---|---|
| Ft. per sec. | Miles per hr. | |
| 0 | 0.0 | 30 |
| 1 | 0.68 | 37 |
| 3 | 2.05 | 58 |
| 5 | 3.40 | 79 |
| 10 | 6.82 | 120 |
| 15 | 10.22 | 150 |

at 30 B.t.u., and $a_2$ at 30 B.t.u. Thus $t_1 - t_2$ and $t_3 - t_4$ will each be

$$\frac{92.27}{30} = 3.076 \ deg.$$

and the wall temperature difference, $t_2 - t_3$, will be

$$26.4 - (2 \times 3.076) = 20.25 \ deg.$$

therefore

$$c = \frac{92.27}{20.25} = 4.556 \ B.t.u.$$

for 2-in. or 9.112 B.t.u. per inch thickness per 24 hr., or 0.38 B.t.u. per hour. In this case $c$ is 30 per cent. larger than $k$.

In the tests on the 4-in. box without a fan, $k$ averaged

$$\frac{1.736 + 1.772}{2} = 1.754 \ B.t.u.$$

Multiplying by $t_1 - t_4$ = 57.8 deg., $H$ = 101.4 B.t.u. Using the same coefficient for inner conductivity as above, namely

$$\frac{9.112}{4 \ in.} = 2.278 \ B.t.u.$$

$t_2 - t_3$ must have been equal to

$$\frac{101.4}{2.278} = 44.51 \ deg.$$

and $t_1 - t_2$ equal to

$$\frac{57.8 - 44.51}{2} = 6.645 \ deg.$$

Thus $a_1$ and $a_2$ equal

$$\frac{101.4}{6.645} = 15.26 \ B.t.u.$$

and

$$\frac{c}{k} = 1.3$$

Table 3, giving the comparison of figures from tests on the same boxes, using air agitation inside, shows a proportionate reduction in the values of $a_1$ and $a_2$ between 2-in. and 4-in. corkboards, the temperature drop at the surfaces of the thick board apparently remaining undiminished.

In agitating the air, $t_1 - t_2$ and $t_3 - t_4$ are decreased, the result being that $k$ approaches the value of $c$; but with a fan it is hardly possible to eliminate entirely the temperature drop at the surfaces. If, with the 2-in. corkboard, the air could be agitated inside and outside with a speed of 15 ft. per second, $t_1 - t_2$ would probably equal

$$\frac{199.1}{150} = 1.327 \ deg.$$

$t_3 - t_4$ also 1.327 deg., making a total of 2.654 deg., which, added to 46.8, makes $t_1 - t_4$ equal to 49.454 deg., and

$$k = \frac{199.1}{49.454} = 4.026 \ B.t.u.$$

Then

$$\frac{c}{k} = \frac{4.253}{4.026} = 1.056$$

Slight differences in the material and treatment of compressed corkboard cause the conductivity to vary from 3 to 5 per cent. either way from the average. Straining for extreme accuracy is therefore useless.

TABLE 3. RESULTS OF TESTS ON CORKBOARD BOXES

| Symbols | 2-in. Corkboard Box | 4-in. Corkboard Box |
|---|---|---|
| $k$ | 3.559 B.t.u. | 1.74 B.t.u. |
| $t_1 - t_4$ | 55.95 deg. | 60.6 deg. |
| $H$ | 199.1 B.t.u. | 105.4 B.t.u. |
| $a_1$ | 79 B.t.u. | 35.15 B.t.u. |
| $t_1 - t_2$ | 2.52 deg. | 3 deg. |
| $a_2$ | 30 B.t.u. | 13.18 B.t.u. |
| $t_3 - t_4$ | 6.64 deg. | 8 deg. |
| $t_2 - t_3$ | 46.8 deg. | 49.54 deg. |
| $c$ | 4.253 B.t.u. | 4.253 / 2 B.t.u. |
| $\frac{c}{k}$ | 1.195 | 1.221 |

Adopting the value of $c$ = 0.38 B.t.u. per hour, and reverting to the formula given in the first article for finding the value of $c$ for a compound wall, the heat transmission through the 20-in. wall of the building per square foot per degree of wall temperature difference is

$$\frac{1}{\frac{16}{5.56}+\frac{2\times\frac{1}{2}}{4.2}+\frac{3}{0.38}}=0.0909\ B.t.u.$$

per hour, or 2.18 B.t.u. per 24 hr.

While the air circulation inside a room may be very feeble, wind might prevail outside. When the air velocity is about 10 ft. per second (a gentle wind) the radiation and convection will increase to 5 B.t.u. per hour per degree difference between the air and wall, and then $k$, the heat transmitted through the 20-in. wall per degree air temperature difference, will be

$$\frac{1}{\frac{1}{5}+\frac{16}{5.56}+\frac{2\times\frac{1}{2}}{4.2}+\frac{3}{0.38}+\frac{1}{1.25}}=\frac{1}{12.01}$$

$$=0.0833\ B.t.u.$$

per hour, or 2 B.t.u. per 24 hr., as against 2.18 B.t.u. when the air was quiet outside as well as inside.

This clearly shows the influence air movement has upon the transmission coefficient $k$, misleading one unless properly defined. On the other hand, the coefficient per degree wall temperature difference is a fixed quantity and is the only proper value to use.

Another common error is that in most tests made in this country the air temperatures only are taken, the resulting coefficient representing $k$ and not $c$.

The heat transmission for 1-in. corkboard, for example, is then simply taken as double that of 2 in., because heat conduction varies inversely as the thickness. However, the latter rule applies only with true conductivity from point to point inside of the material, not when referred to temperatures obtaining outside of the test piece. For instance, with the above 2-in. corkboard, the heat actually transmitted through 1-in. corkboard per degree air-temperature difference, assuming 25 deg. wall temperature difference, is

$$25\times 9.13=228.25\ B.t.u.$$

per 24 hr. Dividing by 30, the coefficient of total radiation with still air, the surface difference is 7.6 deg., and the whole air-temperature difference $t_1 - t_4$ is

$$7.6+25+7.6=40.2\ deg.$$

Hence the heat transmission actually desired (for instance, when computing temperature curves) is

$$\frac{228.25}{40.2}=5.68\ B.t.u.$$

not 2 × 3.5 or 7 B.t.u. However, $c$, figured here at 9.1 B.t.u. for 1-in. thickness is the value actually required.

Having determined the heat transmission through the walls—namely 2 B.t.u. per square foot per 24 hr., or 0.0833 per hour per degree air-temperature difference—calculations can be made as to their neutralizing effect. Evidently the outer wall surface will average only a trifle cooler than 79.6 deg., the average air temperature outdoors at 7:52 a.m. If the refrigerating machine has been kept running until then, the curve of temperature through the walls will be as in Fig. 1. The heat transmission per square foot per hour will be

$$(79.6-32)\times 0.0833=3.965\ B.t.u.$$

The outside surface-temperature drop

$$(a-b)=\frac{3.965}{5}=0.793\ deg.$$

The inside surface-temperature drop

$$(f-g)=\frac{3.965}{1.25}=3.172\ deg.$$

Thus $b$ will be

$$79.6-0.793=78.807\ deg.$$

The points $c$, $d$ and $e$ can be obtained

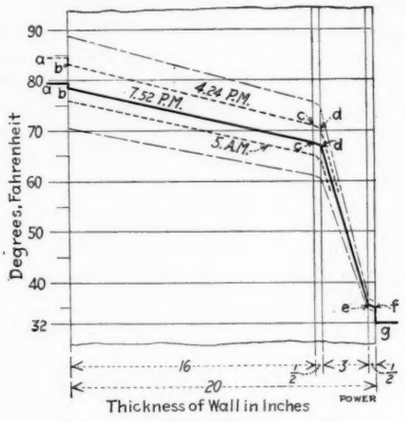



FIG. 1. TEMPERATURE CURVES

from the inner heat conducted through the materials as follows:

The flow of heat through each layer being 3.965 B.t.u. per hour,

$$c=78.807-\frac{3.965}{\frac{5.56}{16}}=67.397\ deg.$$

$$d=67.397-\frac{3.965}{\frac{4.2}{\frac{1}{2}}}=66.925\ deg.$$

$$e=66.925-\frac{3.965}{\frac{0.38}{3}}=35.705\ deg.$$

$$f=32+3.172=35.172\ deg.$$

From these data the mean temperature of the several layers comprising the wall is, for the 16-in. brick wall, 73.102 deg.; first ½ in. of cement, 67.161 deg.; 3-in. corkboard, 51.315 deg.; second ½ in. of cement, 35.438 deg.

The purpose here in calculating intermediate temperatures to an accuracy of several decimals is to have the various temperature drops agree in total with the entire wall-temperature difference. In actual measurements, probably none of the readings would come within 1 deg. of the figures given.

The table of hourly air temperatures given in the previous article shows an increase as high as 90 deg. during the day, which would seem to add materially to the heat transmission. It is therefore interesting to investigate the influence of this higher temperature upon the temperature curve of the wall. The average air temperature between 7:52 a.m. and 7:24 p.m. is obtained at 4:24 p.m. and is 84.2 deg. At the latter temperature the heat flow per hour will be, say

$$(84.2-32)\times 0.0833=4.348\ B.t.u.$$

per square foot. The dotted line in Fig. 1 labeled 4:24 p.m. represents the curve of temperatures obtaining under this condition, namely: $a$ = 84.2 deg., $b$ = 83.33 deg., $c$ = 70.82 deg., $d$ = 70.302 deg., $e$ = 35.985 deg., $f$ = 35.48 deg., $g$ = 32 deg.

The mean temperature of the several layers will be, for the 16-in. brick wall, 77.075 deg.; first ½ in. of cement, 70.561 deg.; 3-in. corkboard, 53.143 deg.; second ½ in. of cement, 35.733 deg.

The amount of heat required to thus increase the temperature of one square foot of insulated wall is:

For 16-in. brick wall,

$$123.5\times\frac{16}{12}\times 0.22=36.2\ B.t.u.,$$

$$B.t.u.$$
$$\times 3.973\ deg.=143.8$$

For first ½ in. of cement,

$$78\times\frac{\frac{1}{2}}{12}\times 0.2=6.5\ B.t.u.,$$

$$\times 3.4\ deg.=22.1$$

For 3-in. corkboard,

$$12\times\frac{3}{12}\times 0.25=0.75\ B.t.u.,$$

$$\times 1.867\ deg.=1.4$$

For second ½ in. of cement,

$$6.5\ B.t.u.\times 0.333\ deg.=2.16$$

Total . . . . . 169.46

The average air temperature of 84.2 deg. occurs at 4:24 p.m.; hence just nine hours were available for increasing the temperature of the wall.

$$\frac{169.46}{9}=18.83\ B.t.u.$$

per hour. In the meantime, heat has entered the room at a rate of 4.348 B.t.u. per hour, or a total of 23.178 B.t.u. per hour. Dividing by 5, the coefficient of heat transfer at the surface, the average difference between air and surface must have been 4.63 deg. and

$$84.2-4.63=79.6\ deg.$$

the air temperature at 7:52 a.m. This calculation shows that when the machine is in operation, the outer-wall temperature does not increase more than

$$83.33-78.81=4.52\ deg.$$

above its normal; therefore the flow of heat into the room during the day cannot exceed

$$(83-35.2)\times 0.0909=4.345\ B.t.u.$$

per square foot per hour.

Before the heat flowing into the room during the night can be determined, the extent of heat abstraction must be known which takes place at the outer surface of the wall, due to the declining air tem-

perature. To determine this quantity accurately involves a rather intricate calculation; it will be close enough to figure again on cooling at the rate of 5 B.t.u. per hour per degree temperature difference between the outer surface and the air. At 7:24 p.m. and 5 a.m. the air temperature averages 74.5 deg. The wall surface during this interval may fall to 76 deg., thus the average difference will have been

$$\frac{83+76}{2} - 74.5 = 5 \text{ deg.}$$

and the heat abstraction

$$5 \times 5 \times 9.6 \text{ hr.} = 240 \text{ B.t.u.}$$

Between 7:24 p.m. and 5 a.m. the heat entering the room is about

$$4 \times 9.6 = 38.4 \text{ B.t.u.}$$

per square foot. Thus there are

$$240 - 38.4 = 201.6 \text{ B.t.u.}$$

to be abstracted from the 20-in. wall. To lower the mean temperature of this wall 1 deg. requires

$$36.2 + 13 + 0.75 = 49.95 \text{ B.t.u.}$$

In abstracting 201.6 B.t.u. from the wall its mean temperature will be lowered by

$$\frac{201.6}{49.95} = 4 \text{ deg.}$$

Fig. 1 gives the temperature curves for 4:24 p.m. and 5 a.m., and the dash-and-dot lines the curves which obtain if there is sufficient heat to raise the outer wall temperature close to 90 deg., and enough heat is abstracted to cool it at night down to almost 70 deg. Accordingly the flow of heat into the room may be taken at 4.345 B.t.u. per square foot per hour during the day, and about 4 B.t.u. per hour during the night. Just at 5 a.m. it is

$$(76 - 35) \times 0.0909 = 3.727 \text{ B.t.u.}$$

A 3-deg. rise in temperature of the room contents is obtained after an inflow of

$$\frac{57{,}800 \text{ B.t.u.}}{1800 \text{ sq.ft.}} = 32.11 \text{ B.t.u.}$$

per square foot of wall surface. During the day a stop could therefore be made for

$$\frac{32.11}{4.345} = 7.39 \text{ hr.}$$

and during the night for about eight hours, neglecting such sources of heat as doors, lights, etc.

Fig. 1 also shows that only very little refrigeration can be stored up when the insulation is placed at the inner or room-side of the brick walls. A 16-hour stop would let the room temperature rise to 38 deg.

By placing the insulation at the outside of the walls, as in Fig. 2, their average temperature is kept quite low and from their proximity to the cold air they influence the temperature rise. The material used is the same as before with a ½-in. finishing coat of portland-cement plaster added on the room side, making the entire wall 20½ in. thick.

The 20½-in. wall might be expected to exert its retarding influence directly refrigeration is discontinued, but while the room contents are colder than the inside wall surface there is no reason for a diminution in the rate of heat supply to the room. How closely these theoretical considerations apply to practice remains to be determined by experiment.

The heat transmission $c$ through this wall per square foot per hour per degree wall-temperature difference equals

$$\frac{1}{\frac{3 \times \frac{1}{2}}{4.2} + \frac{3}{0.38} + \frac{16}{5.56}} \quad \frac{1}{1 \quad .125}$$

$$= 0.0899 \text{ B.t.u.}$$

and $k$ per degree air-temperature difference, still assuming wind on the outside of the building, equals

$$\frac{1}{\frac{1}{5} + 11.125 + \frac{1}{1.25}} = \frac{1}{12.125} = 0.0825 \text{ B.t.u.}$$

This is 1.98 B.t.u. per 24 hr., as against 2 B.t.u. with the 20-in. wall. The air-temperature difference is

$$79.6 - 32 = 47.6 \text{ deg.}$$

and the wall-temperature difference

$$47.6 \times \frac{0.0825}{0.0899} = 43.68 \text{ deg.}$$

the difference between the two being 3.92 deg., one-fifth of which, or 0.784 deg.,

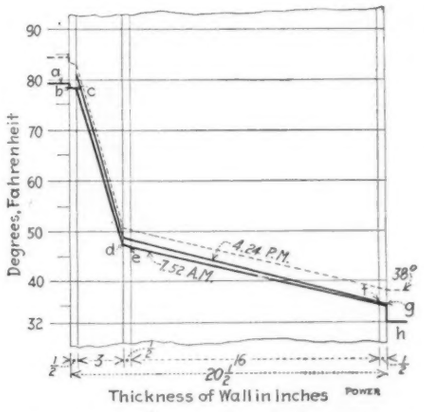



FIG. 2. INSULATION PLACED OUTSIDE OF WALLS

is outside of the wall, and $\frac{1}{1.25}$ or 3.136 deg., inside. The lowest temperature curve in Fig. 2 can now be readily computed: $a = 79.6$ deg., $b = 78.816$ deg., $c = 78.348$ deg., $d = 47.354$ deg., $e = 46.886$ deg., $f = 35.586$ deg., $g = 35.136$ deg. and $h = 32$ deg.

This curve is supposed to apply at 7:52 a.m. The total heat transmission is then equal to

$$(79.6 - 32) \times 0.0825 = 3.927 \text{ B.t.u.}$$

per hour.

To show how little the temperature of the brick wall is now affected by temperature changes outdoors, a second curve is drawn in representing the conditions obtaining at 4:24 p.m., the end of the warming period. In this curve the heat transmission is

$$(84.2 - 32) \times 0.0825 = 4.306 \text{ B.t.u.}$$

per hour.

Even with this construction a 3-deg. temperature rise in the room would be reached after a stop of eight hours, because the brick wall only comes into play when the inner wall temperature rises. When the machine is in operation the room is normally at 32 deg. and the wall surface at 35.14 deg. It has been shown that with the 20-in. wall the room temperature would be 38 deg. after a stop of 16 hours. The question is, how many hours can a plant be shut down with this 20½-in. wall before the room is up to 38 deg.? The dotted line in Fig. 2 shows a curve beginning with the maximum outer surface temperature of 83.34 deg. and ending with 38 deg. inner surface temperature, the heat transmission being then

$$(83.34 - 38) \times 0.0899 = 4.076 \text{ B.t.u.}$$

per hour.

To reach 38 deg., which means a 6-deg. rise,

$$32.12 \times \tfrac{6}{3} = 64.24 \text{ B.t.u.}$$

must enter the room. On the basis of 4 B.t.u. per hour, this takes

$$\frac{64.24}{4} = 16.06 \text{ hr.}$$

plus the time required to raise the wall temperature over the curve of 7:52 a.m. The difference in mean temperatures of the two curves for the inner 17 in. of wall thickness is 3.092 deg. Multiplying by 49.2 B.t.u., the heat capacity of this portion of the wall, 152 B.t.u. is found to be required to thus heat up the wall.

As the heat is being supplied only at 4 B.t.u. per hour, it evidently takes

$$\frac{152}{4} = 38 \text{ hr.}$$

additional or a total of 54 hr. before the room temperature is up to 38 deg. The 38-hr. gain over the other construction is entirely due to placing the walls within the cold zone instead of, as usual, out in the warm zone where they can do no good. With a thinner brick wall the time would be reduced. No saving is made in refrigeration, for as soon as the machine is started again the heat accumulated must be removed and the temperature of 32 deg. regained, but in many plants a shutdown, weekly or daily, would be a great convenience, dispensing with the third or even the second shift of operators.

A plain solid wall, the heavier the better, erected at the inside of the cold room, preferably using the building walls themselves by leaving the insulation outside, is the most rational expedient for accumulating refrigeration. Hollow tile is not recommended because its heat capacity per cubic foot is less than for a solid wall, and the air spaces are of doubtful insulating value because as

built they usually form communicating air channels likely to gather moisture. Their ability to resist fire also leaves much to be desired.

Exterior insulation is still novel in this country, but deserves consideration. Aside from the valuable feature of accumulating refrigeration by means of the cold walls, as explained, the overall insulating effect will be improved because the wall is protected from wind, rain and frost, a damp wall conducting heat readily. Further advantages are decreased wear and tear of building, protection in case of fire, especially on the roof, because modern insulating materials do not support combustion.

The advantage of continuity incidental to interior insulation might be urged as an argument against exterior insulation, but when the insulation on a wall, for example, is carried up 2 ft. or more above the interior ceiling line the same effect is obtained because the heat is then separated from the interior by that amount of brick. On the other hand, the steel framework conducts a great deal of heat into the building, sometimes piercing the insulation at every beam, column and floor, while exterior insulation could be erected without a single break. It could be applied to most old and new buildings, and when laid in the forms of new concrete buildings the cost should be much lower than that of interior application.

In the next article, the application of brine-storage tanks and the piping therefor will be gone into, these being in common use and necessary where there are no other means for accumulating refrigeration.

## Troubles in an Absorption System and Remedies

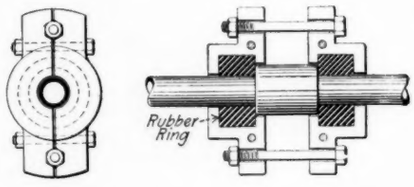
By W. S. Luckenbach

In the issue of July 30, some of the troubles that may perplex those handling an absorption refrigeration system were discussed. The following takes up other troubles and their remedies:

### Failing Capacity

It may happen that a machine has been making its full capacity, but is gradually failing, although it is apparently working as usual with the rich and poor liquor the same, no leaks and the temperature of the cooling and distilled water the same. To locate the trouble and apply a remedy several points must be considered. There may be more lids off the freezing tank than usual, thereby exposing more ice cans to the warm atmosphere, or the temperature on top of the bath may have increased, due to additional warm air from the machine room or elsewhere. The liquid ammonia pipe may be subjected to additional heat that would not be detected unless the pipe had a thermometer close to the expansion valve. Referring to a standard table of properties of saturated ammonia gas it will be found that a difference in the increase of temperature from 70 to 75 deg. F. decreases its heat-absorbing power nearly six times.

The importance of maintaining as low a temperature as possible of the liquid anhydrous ammonia at the expansion valve cannot be overestimated, and the writer would no more think of running without a thermometer inserted near the expansion valve than to discard the thermometer in the brine tank. For instance, if certain results are obtained with a certain temperature at the expansion valve and the temperature increases, say 5 deg. (which no one can detect by simply grasping the pipe) the thermometer will show it at once. Otherwise one might wait until the brine had gone up several points and then wonder what was the cause. With the thermometer one would know at a glance that probably either the condensing and cooling coils needed cleaning or needed more water, or, if neither of these reasons accounted for the decrease in capacity, then the proper amount of liquid ammonia was not passing the expansion valve, or moisture had gotten into it by the liquor having been too high in the generator.




Split Flange Collar for Stopping Leaky Threads

The presence of moisture can be readily ascertained by drawing some of the ammonia into a piece of 1-in. pipe about 1 ft. long capped at one end and having a handle securely attached so as to hold it off some distance when drawing. A little poured into a shallow vessel to evaporate, will show by the amount of water left the proportion of moisture. If the liquid ammonia was originally dry it will become so again by carrying the liquor in the generator a trifle low for a day or two.

Another cause for an ammonia pump not pumping its full capacity is its becoming partly gas bound. To correct this it is well to have a connection from the poor liquor coil at the absorber to the pump suction pipe; then, by partially closing the valve on the suction pipe at the absorber and opening the poor liquor connection wide, the poor liquor passing into the pump will absorb all the surplus gas in a few minutes. Too much brine agitation will increase its temperature. Enough agitation to keep a fairly even temperature is all that is required; more than that is a loss.

### Decreasing Liquor Density

Another perplexing thing is to have the densities of both the rich and poor liquors decrease without any apparent cause. If the loss amounts to, say, 1 deg. in a few days, without any odor being noticeable, gas is evidently escaping through leaks in the pipes or joints of either the condensing, cooling or absorber coils or pipes, and as the cooling water passes over, or through them, it absorbs the escaping gas, so that the latter is not detected unless the water is tested with the usual solution for that purpose.

If the condensing coils are of the atmospheric type, the leak can be easily located by a whitish substance which in a few days will form a crust. It is plain that these leaks must be stopped, and without shutting down. The illustration shows an excellent split-flange collar, recessed for two pure gum rings ¼ in. thick, bolted to a pipe coupling for stopping leaky threads. It can also be used on any fitting or return bend. The flanges are simply rough cast iron with no machine work except drilling one bolt-hole; the other one is cored. The writer has used these for years, always successfully. It is well to keep on hand a few for the various pipe sizes, in case of an emergency.

### Importance of Temperature Variations

A 5-deg. variation in temperature of the distilled water has more effect in the production of ice than may at first be realized. Raising or lowering the temperature 1 deg. means adding or abstracting 1 B.t.u. for each pound of water. Hence, by lowering the temperature 5 deg. in a 50-ton machine (2000 lb. per ton)

$$2000 \times 5 \times 50 = 500{,}000 \text{ B.t.u.}$$

of heat less need be extracted by the expanded gas. This is equivalent to about one ton of ice per 24 hours on a 50-ton machine, allowing also for meltage in removing the ice from the cans. This again demonstrates the importance of paying attention to seemingly unimportant details.

## Concerning Tank Paints

The writer would be glad if some of the readers of this department who have had experience along the lines indicated, would answer the following questions:

Are paints containing an excessive amount of oil deleterious to iron coils used in refrigerating work, or to tanks containing salt or calcium chloride brine, or to water tanks?

Is black asphaltum paint the best or the only good one to use on new coils, or coils that have been in the salt brine 12 years and then cleaned to be used in calcium chloride brine?

B. A. Warren.

Rochester, N. Y.

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# POWER

## Superheated Steam

In our issue of July 23 is a resumé by Professor Dwelshauvers-Dery of a remarkable treatise by his friend and pupil, Dr. Armand Duchesne. Professor Dwelshauvers-Dery is professor emeritus at the University of Liége and founder of its steam-engineering laboratory where Dr. Duchesne's experiments which form the basis of the treatise were made.

One point brought out by the experiment is of special interest. The temperature of steam was measured simultaneously by two mercury thermometers, one with its bulb immersed in the steam itself, the steam passing out through a stuffing-box, one in a thermometer cup filled with oil, and by a very sensitive thermocouple. The current of steam flowed very sluggishly past the temperature-measuring devices in the order named, and when the steam was saturated their indications were substantially the same. But when the steam was superheated, a very considerable difference developed between the mercury thermometers, especially that in the oil well, and the thermocouple, amounting in some cases to over 80 deg. C., or about 150 deg. F.

The thermocouple is very sensitive to impact, and it is difficult to get with it reliable indications of the temperature of the mass in which it is immerged if that mass is in rapid motion. Efforts to explore, for temperature, the axis of a jet of expanding air and steam with a thermocouple consisting of a fine wire stretched through the jet axially, resulted in the indication of 80 or 90 deg. for the final temperature, when by the relation of pressures and temperatures with adiabatic expansion the temperatures should be below zero, and the jet showed by refraction the presence of ice particles. In this case the junction was surrounded by steam flowing with a high velocity, and its impact and friction evidently produced an indication of a temperature certainly above that of the mass. But in the Duchesne experiments the couple was in a current of steam moving very sluggishly and hardly subject to serious disturbance from impact. Furthermore, its indications coincided with those of the thermometer and with the temperatures given by the steam tables for saturated steam of the observed pressure.

Why did it read so much higher than the thermometer for superheated steam?

The reason given by Dr. Duchesne, and it is interesting as bearing upon the behavior and usefulness of superheated steam, is that while the steam is saturated its temperature is the same throughout. It cannot get any higher than that due to its pressure, and if it gets any less the steam will condense and more steam of the fixed temperature will take its place. The temperature of the mass immediately surrounding the thermometer and the thermocouple is in this case the same, and the readings are practically identical.

Superheated steam, however, can cool off without condensing, and is a very poor conductor. A film of such steam about the thermometer cup becomes cooled by transmitting some of its heat to the metal. It clings around the cup and the thermometer records its temperature and not that of the hotter mass. The thermocouple was made of wires as fine as hair and of so little heat capacity that they were not similarly affected.

This explanation furnishes a reason why superheated steam, notwithstanding its greater temperature, loses less by radiation when passing through a steam main than does saturated steam. It appears to be a well authenticated fact that less heat will be radiated from a pipe filled with superheated steam at a high temperature than from one filled with saturated steam at a considerably lower temperature. The explanation also accounts for the well known reluctance of superheated steam to give up its heat to the cylinder walls, whereas the instant the steam becomes saturated the heat exchange is prompt and vigorous. Many of the experiments upon superheated steam have been made with mercury thermometers. It will be interesting to see

what the effect of this phenomenon has been upon their results, and whether its exploitation may not explain some of the inconsistencies which exist between the results of different experimenters.

## Value of a Sketch

In commenting on an article which appeared in the July 19 issue under the heading "Preparing for License Examination," a contributor in this issue, page 236, adds a suggestion relative to the value of a sketch.

Ability to convey ideas by making a sketch is very valuable to an engineer, and the importance of acquiring this art cannot be too strongly emphasized. Some are fortunate enough to be gifted with this ability; others acquire it only by tedious effort. Most of us are in the latter class. This is because sketching is, contrary to the general belief, difficult, unless it is a natural gift. It is a fact that the majority of highly proficient draftsmen cannot make good free-hand drawings and yet can produce very accurate work with tools.

As an instance of where a sketch was valuable, the following is interesting:

A German engineer unable to speak English was being examined by an American board unable to speak German, so the board and the applicant could not understand each other. The German, a technical graduate familiar with the proper procedure of examinations, conceived the idea of showing what he knew of boilers, design of joints, furnace construction, and engine and valve design, etc., by sketches made on a blackboard. He received a license without further questioning because he proved his familiarity with power-plant equipment by his ability to sketch it.

To the young engineer the sketching of things about the plant is one of the best means of acquiring a lasting knowledge of the things which he operates. In examining the object to be sketched he discovers details that otherwise would have escaped attention.

The value of the sketch does not end here. As the contributor suggests, a sketch will often tell what words fail to convey. But if one finds it difficult to express himself in words there is no better way of overcoming that trouble than by drawing sketches. To produce an object on paper he must first study it analytically. Unconsciously he acquires the coveted attribute of being logical and soon finds that his mind can direct his tongue as well as it guides his hand.

## One Schedule is Wanted

A comparison of the two standards of flanged fittings and flanges is made on other pages of this issue and a statement issued by the manufacturers giving their reasons for not adopting the "1912 U. S. Standard" is also published. A study of the article first mentioned will show the exact differences between the two and furnish a basis on which to judge the contribution supplied by the manufacturers.

For the standard fittings it may be noticed that most of the variations occur in the dimensions for long-turn elbows and those for reducing fittings. The societies' schedule maintains uniform face-to-face dimensions for both the straight and reducing fittings of a particular diameter, while the manufacturers make the length of the reducing fitting the same as the old standard, and it is thus shorter than the straight fitting. With the manufacturers' new standard the fittings now in a plant may be replaced with no change in the piping. This, of course, would mean a temporary convenience for the consumer, but in the long run would it not be of greater convenience to be able to interchange straight and reducing fittings, or *vice versa,* as the case required? Apparently this is one of the strong points of the "1912 U. S. Standard."

There is some difference between the two standards on Y-fittings and reducers, and in the extra-heavy fittings and flanges the dissimilarities are the same as those mentioned for fittings of standard weight. For extra-heavy flanges for piping nine inches in diameter and above, the societies increase the bolting beyond the requirements of German or English practice with which the manufacturers' new standard closely agrees. In the opinion of the manufacturers the increase is excessive, and on the larger flanges the bolts are so large as to be cumbersome and inconvenient from the steamfitters' standpoint.

**Convenience is a factor that should be considered, but above everything else is safety. With any doubt concerning the strength of the bolting it is better to err** on the safe side. Both parties no doubt have good reasons for the adoption of the individual schedules, but the differences are not so great as to prohibit settlement by a joint committee. Personal opinion or prejudice should not influence the issue, nor should *temporary* expense to the manufacturer or inconvenience to the consumer be given primal importance. One schedule, and only one, with convenience giving first place to safety, is wanted.

A joint committee is the solution. A little give and take on both sides would settle all differences. It is up to the society committee and the manufacturers to throw aside all personal considerations, to get together, and to agree on a schedule that will last as long as the present materials and pressures are employed.

## The Massachusetts Formula for Circular Segments

There has been considerable speculation as to the derivation of the formula prescribed by the Massachusetts Board of Boiler Rules for the areas of circular segments. It is said that the chief inspector dug it up out of an old note book, but that neither he nor anybody else knew how it happened nor why it gave the right answer. Some of the applicants for license have been asked lately how this formula was derived, possibly as a test of their frankness. The derivation is explained by Prof. Marshall, of the Sheffield Scientific School, on page 240.

How about that heating system? Don't be like the man who couldn't shingle his roof when it rained and didn't shingle it when it didn't rain because it didn't need it. Now is the time, when it is out of commission, to comb out of the system all of those kinks and snarls which unattended to will make the winter's work harder and the service less satisfactory.

If *The Engineer* had given Professor Christie credit for the sage remarks on European turbine practice in relation to bearings given in its issue of July 26, page 93, it would not be quite so responsible for a misstatement made. Turbine-bearing surface speeds of 60 ft. per second (not minute) are common practice. The same mistake appears in the original paper by the professor.

# Readers with Something to Say

**A letter good enough to print will be paid for. Ideas, not mere words, wanted**

## Providing for Pipe Expansion

An engineer was piping a large vertical boiler from which a long pipe was designed to discharge steam into a header. When this pipe was ready to make the final connection it was 3 in. too short. He turned steam into it, and when heated it was just the right length. This shows how a pipe expands, but when this pipe is cooled, it will make a strong effort to regain its former position. Unless changing lengths are provided for, a pipe line is sure to be subjected to heavy strains, that may cause ruptures.

Another engineer was laying a long line of pipe, the end of which was 7 in. from a brick wall. When steam had heated this pipe he was surprised to find an ell on the end of it, hard against this wall. When cool it went back to its original position.

Fig. 1 illustrates the main steam pipe of a plant, showing where it comes through a brick wall and enters a separator just above the throttle valve. Four ells were used and ample provision was thus made for expansion, but its appearance is very crude and unsatisfactory. The main part coming through the wall might have been continued as shown by dotted lines, then brought forward and turned downward to connect with the separator shown. Two ells would have sufficed and expansion would do no harm, as the pipe could spring enough to compensate for it.

Fig. 2 is the exhaust-steam piping from the cylinders of a double engine. The piping, as will be seen, is so arranged that the pipe from each cylinder can expand without damaging the other. The Y-fitting shown in connection with a 45-deg. ell is recommended because it allows exhaust steam to escape freely, with less friction and back pressure than if a common tee was used. Gate valves allow either exhaust pipe line to be shut off, if one cylinder is not used.

Fig. 3 illustrates a section of piping in another plant which has proved satisfactory. The upper pipe is carried straight through, as there is ample room for it to expand. This expansion does no harm to the branch lines, as they are long and free to move. In the lower pipe the expansion must be divided, hence the ells and extra piping used cause steam to follow an indirect course where it passes through the brick wall shown, but this arrangement admits of expansion without causing leaks or other evidence of excessive strains.

Fig. 4 shows the provision made for expansion in a long line of 8-in. pipe, carrying a light pressure. Two long sweep bends are used to carry the pipe 10 ft. out of a straight line, two more bring it into line again. This constitutes a 10-ft. offset, and the pipe extends beyond these points in both directions. It works well in practice, as no leaks have developed in 10 years' service.

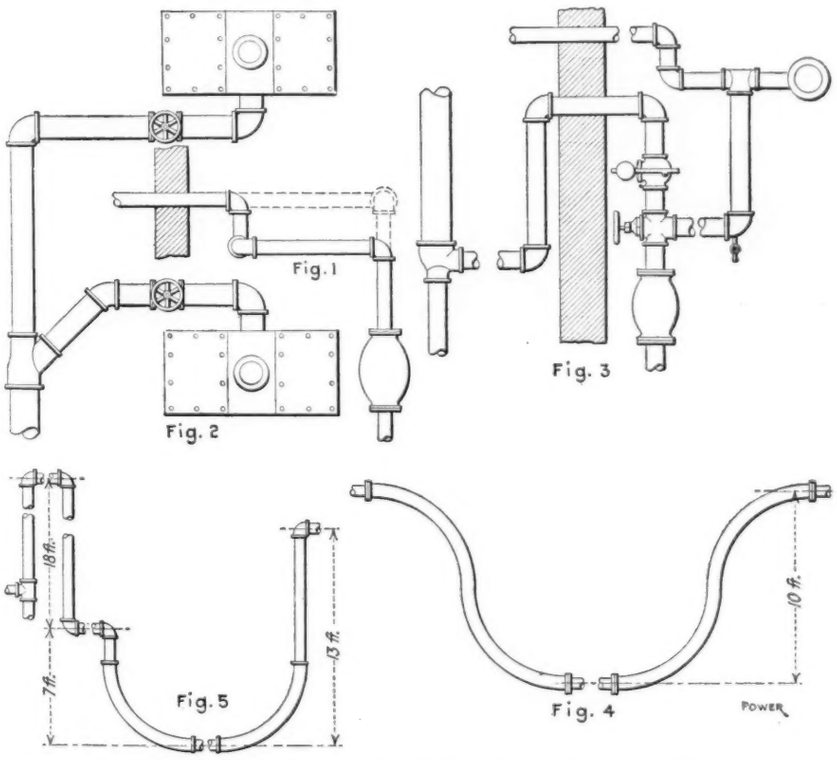
Fig. 5 shows a line of 10-in. pipe which might have been carried in a straight line (as indicated by dotted lines), and fitted with an expansion joint to provide for variations in its length, but it was decided to adopt offsets instead.

Starting at the right, an ell followed by a piece of pipe carries it 8 ft. out of line, then a long sweep bend gives 5 ft. more. Then come several lengths of straight pipe, followed by another bend and a short pipe which carry it back 7 ft. A right-angle turn and more pipe follow, until there was a suitable place for another ell. The next length is 18 ft., then it is extended 14 ft., followed by a turn at right angles to the main line, which gives ample chance for expansion without causing leaks.

This arrangement has given satisfactory service, hence the principles involved are recommended for application to similar conditions. The piping shown passes through four brick walls. In all such cases the pipe should rest on substantial rollers well anchored, so as to move easily when expanding and contracting.

W. H. WAKEMAN.

New Haven, Conn.




STEAM-PIPE CONNECTIONS, SHOWING PROVISION FOR EXPANSION

## The Massachusetts License Law

The new clause in the Massachusetts license law, that went into effect Jan. 1, 1912, seems to have filled a long felt want for men seeking a second- or first-class license.

Applicants are now sure of an impartial examination with three of the examiners to pass judgment. Under the old system there was only one examiner which dissatisfied many engineers with the result of their examination. With the new order of doing things, the applicants say that they are satisfied with the examina-

tion. Some making this remark did not receive any license, and in one case the applicant was one of the "hard knockers" of the old administration.

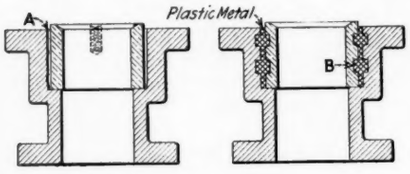
Does not this show that the engineers are pleased with the new law?

W. L. NESBIT.

Springfield, Mass.

## Repairs with Plastic Metal

For many repair jobs about the plant, plastic metal* is useful. It will run into



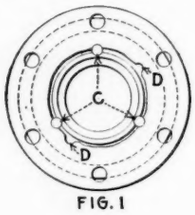




FIG. 1

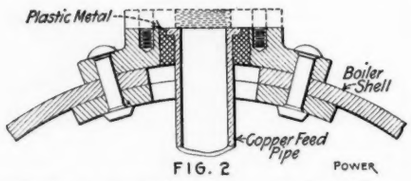



FIG. 2

SHOWING CHECK VALVE AND REINFORCING FLANGE REPAIRED WITH PLASTIC METAL

almost any shape at a low heat and is hard and tough enough to hold under high pressure. For lining bearings, thrust collars, bushings, glands, coating plungers, etc., it has few equals. Price is the drawback, but where time is money the metal is cheap at any cost.

It helped me out in several cases where the job seemed hopeless, as the following instances will show: The boiler-feed check-valve seats had been loose for some time, but no stop could be made for overhauling them, until the body of the castings had worn as at *A*, Fig. 1. The worn seat was turned, as shown at *B*, and holes drilled and tapped vertically in the casing and seat to admit screw pins *C* for reinforcement. The seat was centered in place and heated, after which the metal was run in through the openings *D*. The metal was then lightly calked. It made a good job and works well under 165 lb. pressure.

The next case was a repair to a nipple for the top feed pipe in a return-tubular boiler. The feed line to the boiler was of ordinary steel pipe, and the down pipe in the boiler was of 2-in. gas pipe connected to a reinforcing flange, which was riveted to the shell, as shown. The whole outfit pitted badly and the outside lengths of pipe were replaced several times. I suspected that the internal feed pipe was in bad condition, and when dismantled it was found that it had been eaten off as if by sulphuric acid.

The solid steel flange was eaten away almost to the stud holes, as shown in Fig. 2. A temporary job was made with a separate false flange (shown dotted) into which a new pipe was screwed. The internal copper pipe tinned and slightly flanged, as shown, was passed into the boiler and held flush with the ring. The metal was then poured in and well calked around the copper pipe, after which the temporary flange was connected. The boiler pressure in the case was 140 lb. and the job has held without any trouble.

J. MCA. HOWDEN.

Melbourne, Australia.

*Any alloy of ordinary fusing temperature sufficiently high to safely withstand the temperature of the steam would do for such work.

## Pressure Chart for Comment

This recording steam-gage chart shows the pressure maintained in our plant for 24 hours. We have one 500-hp. and one 250-hp. water-tube boilers, hand fired with buckwheat mixed with duff* coal of which about 2300 lb. are burned per hour. Forced draft of 0.3 in. of water is maintained, and a damper regulator is also provided. The fires are cleaned three times during a 24-hour run.

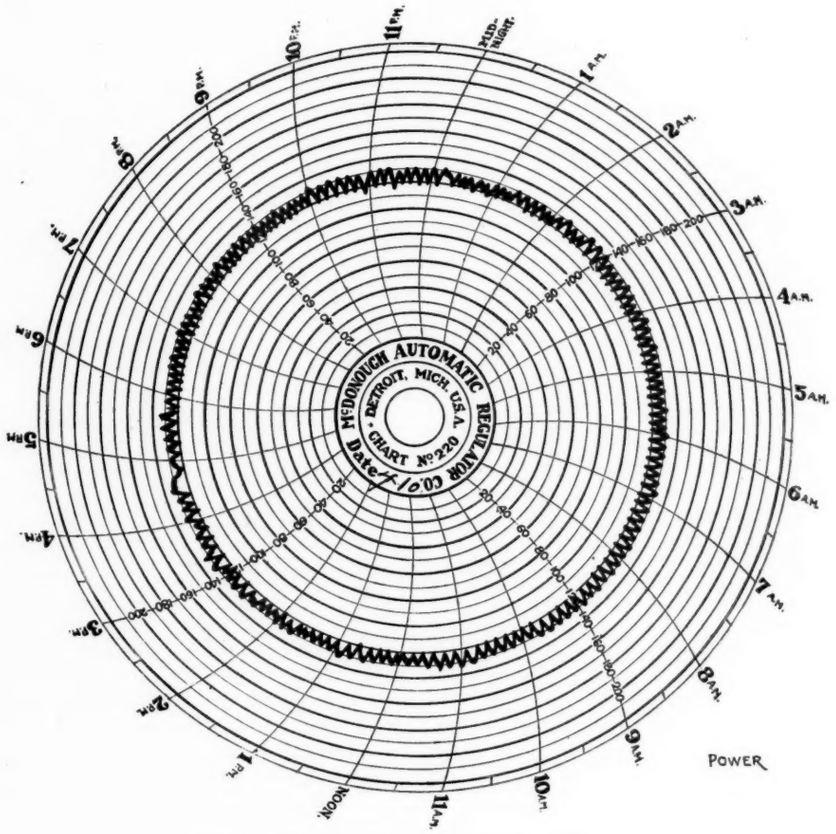
Until recently the boilers had automatic stokers which were dispensed with on account of poor economy and other troubles. We now burn nearly 20 per cent. less coal, keep the furnace walls in better condition and obtain a more uniform steam pressure than with the stokers. The load carried is the same as when the stokers were in use.

I would appreciate comment on the chart and would like to see more about results with stokers and comparisons of charts obtained with automatic and hand-fired furnaces.

BURT M. SEYMOUR.

Delphos, Ohio.

[*"Duff" is a fine coal, the name being given to washed bituminous slack.—EDITOR.]




STEAM CHART FROM HAND-FIRED BOILERS

## Rack for Pipe Fittings

The pipe-fitting rack illustrated is designed to take the place of those which consist of nail kegs, barrels, old boxes and very often floor space. The design may not be new, but the rack has proved very valuable in our plant. It consists of a number of spaces arranged in rows and labeled for the kinds of fittings which are to be hung in each division. Instead of the usual rows of boxes or bins, nails

are driven and the fittings hung on them

It can be cheaply constructed, and will not take up much room. It has all the desirable features that go to make for simplicity and time saving, when properly arranged and a place for each fitting designated. Generally provision for ⅛ to 2½-in. fittings will be found sufficient. All of the fittings are out in plain view, which is also advantageous.

Dirt does not accumulate in this rack as it does in boxes, and no boards are needed to build it. Any wooden wall will answer, but if appearance is a consideration, a smooth board facing can be added if needed and painted to make it more attractive. With smaller nails for small fittings driven close together, the maximum capacity for a given space can be realized. Care should be taken in driving nails for hanging ells and tees and two nails instead of one should be used for each fitting.

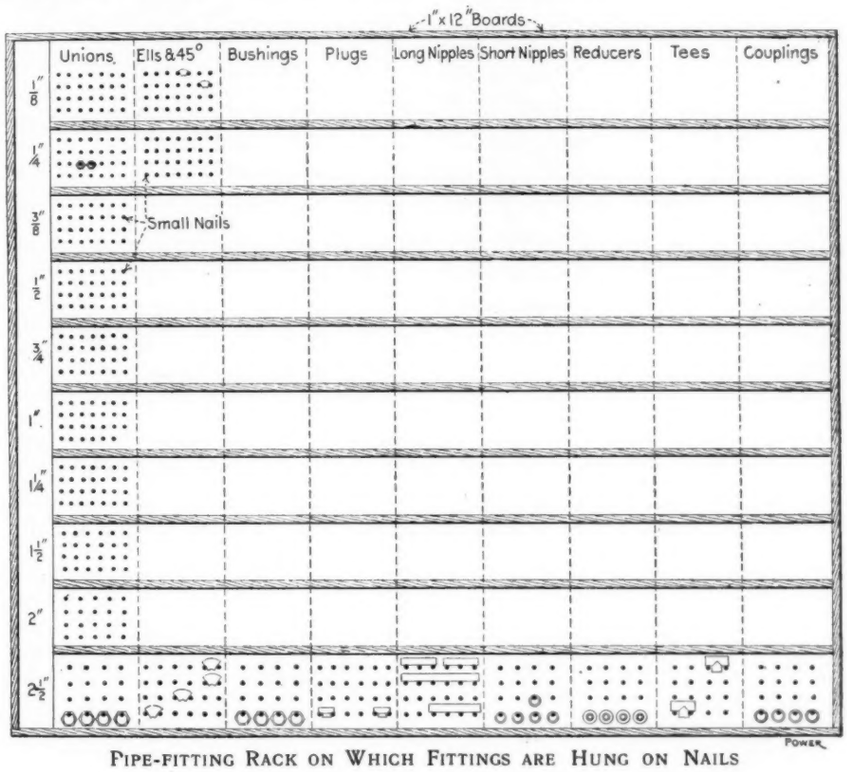
Other fittings, such as short nipples up to 3 in. long, can be hung on single nails driven slightly slanting. For longer nipples the nails should be in a row close together, as illustrated.

It is well not to allow odds and ends of pipe with fittings attached to lay around, but to strip all that are useful and hang them on the rack, throwing the others into a barrel or box kept for that purpose. It is a bad habit to save everything from a damaged flywheel to a broken hacksaw blade, thinking that they may be "good for something." Junk is junk and should be treated as such by selling it when it accumulates. In this way the "good for something" is realized to the best advantage.

The orderly storing of stock should be carried farther by arranging labeled boxes, or bins to contain screws, bolts, washers, nuts, valve wheels, disks, etc.

JOHN J. RAMSEY.

Wellington, Kan.




PIPE-FITTING RACK ON WHICH FITTINGS ARE HUNG ON NAILS

## Sawdust Stops Belt Slipping

Our air compressors are belted to motors, the pulleys of which are of paper. These compressors would always give trouble when the riveting floor was full of bolted-up material to be riveted. The belts would slip, and down would go the air pressure until it was impossible to drive a tight rivet. Rosin and a dozen kinds of belt dressing were tried, but they only seemed to work while they were being put on. It was also difficult to apply dressing on account of the location of the belts.

This trouble became so great that we decided to coat the pulleys with anything that would correct the slippage. The sawdust under a crosscut saw in the template room looked good so we decided to try it. Glue was suggested as a sticking medium and Saturday afternoon as the time, as it would set over Sunday.

The motor belt was taken off and the machine turned over by hand until an even coat of glue covered the entire face of the pulley. A pan full of sawdust was held under the pulley, which was kept turning until its entire surface was covered. The result was gratifying as all the next week the machines served by that compressor kept going, while the other compressor "fell down" as usual. In fact, the sawdust did so well that a couple of extra hammers were supplied by the compressor without reducing the pressure.

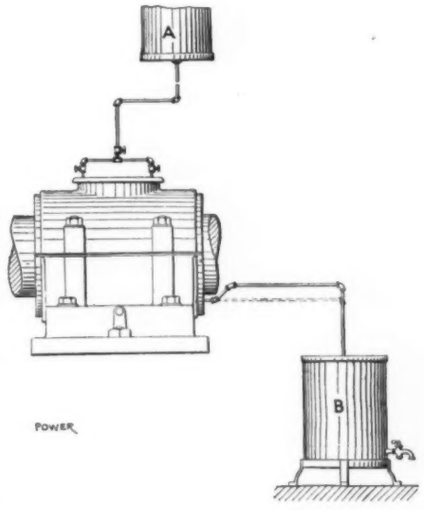
The next Saturday the other compressor-motor pulley was sawdusted, and the riveter boss who had been kicking for more air wanted to know what happened. Not only was it an easy and cheap way to correct the complaint, but the cost of power to run the compressors with the slippage eliminated, was less than it had been.

A. E. DIXON.

Cleveland, Ohio.

## Lubricating a Large Bearing

The illustration shows a ring-oiling bearing for a 6 1/16-in. shaft driven from a 500-hp. engine by a 48-in. belt. The distance between centers is very short, requiring a very tight belt to develop the required power. The tight belt caused excessive friction, resulting in hot boxes. Oil was fed to the bearings in a continuous stream from an elevated tank *A*. The end near the bottom of the box was tapped for a ⅜-in. connection, which was piped to a can *B*, as shown by the dotted lines. This arrangement kept the oil recess in the bottom of the box from filling, as the oil coming in at the top would run right through and out at the bottom.




ARRANGEMENT FOR OILING LARGE BEARING

After having considerable trouble with heated bearings it was suggested that the pipe be raised, as shown by the full lines, allowing about 1½ in. of oil in the bottom of the box before it would run out, thus giving the rings a deeper pool of oil to turn in. This furnished sufficient lubrication, as no further trouble was experienced.

J. W. DICKSON.

Memphis, Tenn.

# Questions Before the House

**Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues**

## Value of a Sketch

To supplement W. Nelson's article on "Preparing for License Examination" in the July 19 issue, I would like to offer a suggestion that seems right in line with his idea. It is that the young engineer should learn early to sketch. He should learn to draw, but he must at least learn to make an intelligible free-hand sketch, and rapidly, too, if he is to succeed in his work. He may never be called upon to produce a carefully finished drawing, but every day, and many times a day, he will meet occasions when a few deft strokes of a pencil will convey ideas, in an instant, which words could hardly have made distinct. A subordinate wants instructions—make him a sketch. The manager wants to know just how you propose overcoming a difficulty—let the lead pencil answer him.

It is not necessary to wait till one reaches a position of responsibility before he reaps any benefits. They come immediately. Every time one makes a sketch of an object, he learns something about that object. Putting the lines on paper makes a lasting impression on the memory.

Begin with something easy—say a pump valve—and work gradually up to more and more difficult things. Try to get the idea with the fewest lines. Try to get in every necessary dimension—and no more. After the sketch is made, put yourself in the other fellows' place and see if you can make the article from the data given.

WILLIAM E. DIXON.

Malden, Mass.

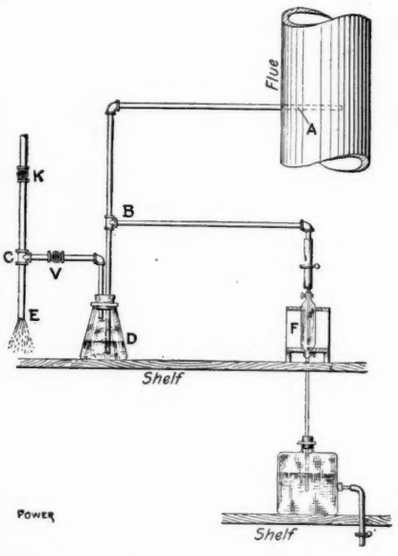
[Editorial reference to this letter is made on page 232.—EDITOR.]

## Flue Gas Sampler

Charles M. Reed's description of a flue-gas sampler, given in the July 16 issue, is a little misleading as to the reason for collecting the gas over mercury, and is in error as to the manner of obtaining the sample.

If it is for immediate analysis, as generally, a sample taken over water, previously saturated with chimney gas, can be measured accurately enough for practical purposes. The gas will be saturated with moisture when the analysis is made, the percentage of moisture depending on the temperature of the gas when analyzed, amounting to 1.16 per cent. at 62 deg. F. and 5.56 per cent. at 92 deg. F., thus reducing the dry gas contained in the sample, and causing a difference in the sum of the oxygen contents.

If the sample is allowed to stand for an hour or so, the $CO_2$ will be materially reduced by absorption in the water over which it is collected. For this reason the mercury sampling bottle as described by Mr. Reed, has been adopted by the Bureau of Mines. This form of sampling was described in a bulletin issued by the Bureau of Mines, and in an abstract by J. C. W. Fraser and E. J. Hoffman, in POWER, Aug. 22, 1911.




PIPE CONNECTIONS FOR FLUE-GAS SAMPLER

Mr. Reed's method of sampling the gas is not practical, for the reason that if steam were allowed to blow through flask *D* the water there contained would soon be evaporated, and there would be only a steam discharge from *E* and no indication of a suction from the flue.

The proper connections are shown herewith. The pipe *A* leading from the flue has one end immersed in the water in the flask *D*, while a branch leads to the sampling bottle *F*. From a tee connection *C* in the steam pipe *K*, a branch is led to *D*, with its end well above the water level. The object of the flask *D* is simply to indicate a proper gas flow, while the steam aspirator is to insure a fresh sample being always drawn from the point of sampling, so that there will be no lag, due to the excess capacity of pipe *A* over that of the bottle *F*.

To operate, regulate valves *K* and *V* so that a moderate agitation is noted in *D*, due to the gas being drawn through the water. When the pipe *A* has been exhausted of all dead gas, open the top cock on the bottle *F*, and regulate the lower cock so that the rate of flow of the mercury will empty the bottle in the time over which the sample is to be taken. The sample is obtained by the suction of the mercury leaving *F*.

O. Z. HOWARD.

New York City.

I wish to thank Mr. Howard for clearing up the reasons for taking flue-gas samples over mercury, and also for calling attention to and correcting the error in the method of obtaining these samples.

While it is true that samples collected over water will absorb moisture, the error in the $CO_2$ content will, as Mr. Howard says, be materially affected. The Bureau of Mines, in the bulletin referred to by Mr. Howard, says in part:

"If water is saturated with furnace gas containing 12 per cent. of carbon dioxide and is then used in collecting a sample from gas having a carbon dioxide content of 8 per cent., the water saturated with gas containing 12 per cent. of carbon dioxide will give up some of the dissolved gas, and the sample will be found to contain more than 8 per cent. of carbon dioxide. The magnitude of the error introduced in this way will vary with conditions, but it will depend largely on the length of the sampling period, and the total time of contact of the sample with the water."

Upon reading Mr. Howard's criticism I made a very thorough examination of a plant to which I have access, and to my chagrin discovered that the connections were made essentially as he has shown them, and not as shown in my drawings. I am very glad that he called attention to the error.

CHARLES M. REED.

Annapolis, Md.

## Taking Steam from Water Column Connections

Mr. Wakeman's letter in the June 18 issue wherein he shows, in Fig. 1, a bad arrangement of water-column connections, reminds me of a similar case. I

was recently called as an expert witness in a court trial to tell what I thought of such an arrangement where, however, the connection was made to the top water-column connection for a stack blower. The boiler was of the portable firebox type and would not generate enough steam without the blower. There is hardly any doubt that the explosion which occurred and killed the fireman and injured two others was caused by this arrangement. At any rate it cost the owners of the boiler about $12,000 to satisfy the courts.

C. R. SUMMERS.

Chattanooga, Tenn.

## Results of Burning Screenings

Mr. Fieux's comment under this heading in the June 18 issue on my article on burning coke screenings greatly interested me, and I will be glad to learn the results he has obtained in detail, if he will furnish them.

If he will refer to the article in question he will find under "Average Pressures and Temperatures" the pressure in the ashpit in inches of water for the first test to be 0.877 and for the second 0.972. These pressures are the actual average of observations taken throughout the tests and are not, as he suggests, the results of computations. The point I wished to make in the paragraph from which he quoted, was that the rate of combustion determines the steam pressure required at the blowers. That is, an increase in the amount of coke fired calls for an increase in the amount of air supplied, which, in turn, since all the air is supplied by the blowers, demands an increased steam pressure at the blowers.

Mr. Fieux states that "this (frequent leveling of the fire) may be good in testing for maximum efficiency, but is extremely poor boiler-room practice"; it is our experience that unless the fire is leveled frequently, a hot fire cannot be maintained. The inconsistency in our experience may be due to the physical properties of the coke itself. While the analysis of the coke burned by Mr. Fieux may be identical with mine, the size may be entirely different. With screenings running as large as ¾ in. in diameter, it is comparatively easy to produce good results. On the other hand, I am anxious to learn how coke that is so fine that at least two-thirds would pass through a $\frac{3}{16}$-in. screen, approximately half through a $\frac{3}{32}$-in. screen, and which contained very little that would go over a ½-in. screen, can be burned without frequent leveling. These figures are merely an estimate, but they show approximately the size of the coke used in the tests. Most of the stuff looks more like gray sand than anything else.

If Mr. Fieux has had experience with coke as fine as this, I would like to learn how his methods of firing differ from those outlined in my article.

The steam used by the blower was obtained by a calibration of the blowers actually used in the tests. This calibration was made by immersing the blowers in a barrel of water on a platform scale and weighing the steam condensed per unit of time for different pressures at the nozzle. From these observations a curve was plotted showing the steam consumed per unit of time corresponding to the different steam pressures at the nozzle. As there was little variation in the steam pressure, which was measured during the course of the test, the steam used was calculated from the point on the curve corresponding to the average pressure at the blowers.

One significant statement is that he "obtained a rate of combustion of 25 lb. per sq.ft. of grate area." He does not say whether with dry or natural coke. Compared with this, in the first test we obtained a rate of combustion of 45.88 lb. per hr. per sq.ft. of grate as fired, or 36.98 lb. per hr. per sq.ft. of grate, and in the second 48.25 lb. per hr. per sq.ft. of grate as fired, or 39.52 lb. per hr. per sq.ft. of grate, dry. This may be the reason he obtained an efficiency 10 per cent. greater than mine.

Another reason why my efficiencies were not higher was that the moisture in the coke in each case was nearly 20 per cent., and no correction was made for this.

In making our calculations, too, the steam used by the blowers in each case was deducted before calculating the efficiencies. I suppose that Mr. Fieux also made this correction.

J. F. MOWAT.

Joliet, Ill.

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The accompanying data of three tests made at various localities with various kinds of coke breeze may interest Mr. Mowat. The worst kind of coke breeze being used is the Solvay, a 72-hr. coke, which is slow burning and generally considered impossible to burn with any efficiency and capacity.

Test of Horizontal Return Tubular Boiler

| | |
|---|---|
| Method of starting and stopping | Alternate |
| Grate surface, sq.ft. | 25.21 |
| Heating surface, sq.ft. | 1321.1 |
| Kind of furnace—Regular—Pin hole grate | |
| Percentage of air space | .9 |
| Kind of coal. Solvay breeze—(fine) 72-hour coke. | |

Total Quantities

| | |
|---|---|
| Duration of trial, hr. | 7.3 |
| Weight of coal as fired, lb. | 6,648.0 |
| Percentage of moisture | 10.49 |
| Total dry coal consumed, lb. | 5,950. |
| Total ash and refuse, lb. | 1,784. |
| Percentage ash and refuse | 29.98 |
| Total water fed to boiler, lb. | 28,358.7 |
| Factor of evaporation | 1.07042 |
| Equivalent evaporation to dry steam, 100 deg. C., lb. | 29,293.2 |

Average Temperature and Pressure

| | |
|---|---|
| Steam pressure, lb. | 96.4 |
| Feed water temperature, deg. F. | 181.5 |
| Stack gases, temperature, deg. F. | 546.8 |
| Ash pit pressure, in. | 1.45 |
| Horsepower developed | 116.3 |
| Percentage of rated horsepower | 93.04 |

Economic Results and Efficiency

| | |
|---|---|
| Water evaporated per pound of coal as fired, lb. | 4.265 |
| Equivalent evaporation per pound of coal, lb. | 4. |
| Equivalent evaporation per pound of dry coal, lb. | 4.922 |
| Equivalent evaporation per pound of combustible, lb | 7.030 |
| Efficiency of boiler, per cent. | 48.16 |
| Efficiency of boiler and grate, per cent. | 38.91 |

Analyses

Coal

| | per cent. |
|---|---|
| Moisture | 10.49 |
| Volatile matter | 3.59 |
| Fixed carbon | 73.97 |
| Ash | 11.97 |
| Sulphur | 0.674 |
| B.t.u. per pound of coal | 10,987.0 |

Test of Babcock & Wilcox boiler

| | |
|---|---|
| Method of Starting and Stopping | Alternate |

Total Quantities

| | |
|---|---|
| Duration of trial, hr. | 10 |
| Weight of coal as fired, lb. | 17,721 |
| Percentage of moisture | 10 |
| Total dry coal consumed, lb. | 14,354 |
| Total ash and refuse, lb. | 2,202 |
| Percentage ash and refuse | 12.4 |
| Total water fed to boiler, lb. | 87,824 |
| Factor of evaporation | 1.2193 |
| Equivalent evaporation to dry steam, lb. | 105,818.5 |

Average Temperature and Pressure

| | |
|---|---|
| Steam pressure, lb. | 119 |
| Feed water temperature, deg.F. | 43 |
| Stack gases, temperature, deg.F. | 503 |
| Steam pressure on blowers, lb. | 75 |
| Ash pit pressure, in. | 0.40 |
| Draft over fire (suction), in. | 0.1 |
| Horsepower developed | 306 |
| Percentage of rated horsepower | 87.5 |

Economic Results and Efficiency

| | |
|---|---|
| Water evaporated per pound of coal as fired, lb. | 6 |
| Equivalent evaporation per pound of coal, lb. | 6.14 |
| Equivalent evaporation per pound of dry coal, lb. | 7.36 |
| Equivalent evaporation per pound of combustible, lb. | 8.42 |
| Efficiency of boiler and grate, per cent. | 53.6 |

Analyses

Coal

| | per cent. |
|---|---|
| Moisture | 19 |
| Volatile matter | 10.12 |
| Fixed carbon | 60.79 |
| Ash | 10.09 |
| B.t.u. per pound of coal | 13,280 |

Test of By-product gas-coke breeze, Sterling Boiler

| | |
|---|---|
| To determine general economy and capacity Method of starting and stopping | alternate |
| Grate surface, sq.ft. | 82.9 |
| Heating surface, sq.ft. | 2,870 |
| Kind of furnace Sterling F.B. arch. Pin hole grates | |
| Percentage of air space, per cent. | 6–7 |
| Kind of coal, Coke Breeze—through $\frac{3}{32}$ in. mesh—about 1 per cent. ½ in. lump. | |

Total Quantities

| | |
|---|---|
| Duration of trial, hr. | 7 |
| Weight of coal as fired, lb. | 14,750 |
| Percentage of moisture | 16.3 |
| Total dry coal consumed, lb. | 12,346 |
| Total ash and refuse, lb. | 2,657 |
| Percentage ash and refuse | 18 |
| Total water fed to boiler, lb. | 90,935 |
| Factor of evaporation | 1.075 |
| Equivalent evaporation to dry steam, lb. | 97,755 |

Average Temperature and Pressure

| | |
|---|---|
| Steam pressure, lb. | 148 |
| Feed water temperature, deg.F. | 187 |
| Stack gases temperature, deg.F. | 473 |
| Typhoon Turbine blowers | |
| Ash pit pressure, in. | 0.764 |
| Horsepower developed | 404 |
| Percentage of rated horsepower | 1.41 |

Economic Results and Efficiency

| | |
|---|---|
| Water evaporated per pound of coal as fired, lb. | 6.16 |
| Equivalent evaporation per pound of coal, lb. | 6.63 |
| Equivalent evaporation per pound of dry coal, lb. | 7.92 |
| Equivalent evaporation per pound combustible, lb. | 10.08 |

Flue Gases

| | |
|---|---|
| Carbon dioxide (Chart) | 13 per cent. |

Mr. Mowat fails to mention the kind of grate and air space used during the test. With a rate of combustion of from 45 to 48 lb. his ashpit pressures of 0.877 and 0.972 in. seem exceptionally low. I believe, however, his error is due to the fact that he has taken the steam pres-

sure at the blower and the rate of combustion of the fire. The thickness of the fire should also be taken into consideration. The only accurate way of taking ashpit pressures is with a U-tube, and not calculating it, as has been done.

Mr. Mowat states that the pressures are the actual averages of observation taken throughout the test. I cannot see how they can be so, for such a remarkable rate of combustion is almost equal to the rate of fine soft coal, and with a poor grade of coke breeze it would be almost impossible. The coke must have burned freely; otherwise it would have been impossible to get the high rate of combustion.

I do not think that the size of the coke breeze enters into the quality of its burning as much as the chemical analysis and the method in which its various constituents are formed. It is well known that a 72-hr. breeze is very hard to burn, whereas an 18-hr. breeze is rather easy. I have seen coke breeze that was impossible to burn without being mixed with soft coal.

I believe, if Mr. Mowat would try a pinhole grate properly designed, he would find that the fire would not need leveling as often as in his previous test.

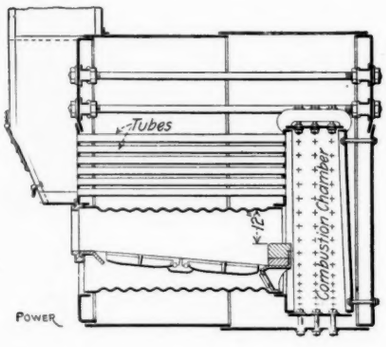
I would like to find out where the discrepancy between our results lies.

ERNEST D. FIEUX.

New York City.

## Unnecessary Risks

Replying to L. A. Suverkrop's comment in the June 25 issue on my letter under this heading in the June 4 issue, I would




SCOTCH MARINE BOILER

state that the expression "crawling on their hands and knees" through the furnace of a Scotch marine boiler to plug a leaky tube without drawing the fire, is not literally true, as Mr. Suverkrop says.

It is not, however, absolutely necessary to draw the fire to get into the "back connection." On account of the sloping grate and fire, even a smothered fire and plank would be below the level of the bridgewall at that end. The bridgewall must be passed in any case, and in ordinary Scotch marine boilers it is 12 in. from the top of the wall to the crown of the furnace, as shown.

CHARLES J. MASON.

Scranton, Penn.

[Unless the fire is exceedingly heavy it is possible to crawl (not on the hands and knees) to the bridgewall on a plank laid over a smothered fire. This would be the worst kind of practice, however, as the gases would, in most cases, overcome the worker. It is questionable if this would ever be necessary and a man would be justified in disobeying an order to crawl in unless the fire was drawn.—EDITOR.]

## Cylinder Condensation

If B. T. Abbott, who asks for information relative to cylinder condensation, in the issue of May 28, will look on page 88 of the catalog of the Buckeye Engine Co., he will find a rule, which, in my experience, has served well in accounting for the missing quantity. It was devised by Mr. Thompson, the designer of the original Buckeye engine, and is as follows:

*12½ to 16½ times the ratio of expansion = missing quantity percentage of indicated.*

The coefficient is to be varied with the range of temperature in the engines, the larger quantity with long ranges and intermediate, down to 12½ for shorter ranges. If good judgment is used in selecting the coefficient, this formula applies well to Corliss and moderate speed engines.

A formula, which takes into consideration nearly all variables, is the one devised by G. R. Bodmer, from experimental data quoted by Kent in his "Mechanical Engineer's Pocket Book," eighth edition, and is as follows:

$$W = C\frac{S\,(T-t)}{L\sqrt{N^2}}$$

in which

$W$ = Weight of water condensed per minute;
$S$ = Clearance surface area in square feet;
$T$ = Mean admission temperature;
$t$ = Mean exhaust temperature;
$L$ = Latent heat of steam at mean admission temperature;
$N$ = Number of revolutions per second.

The coefficient $C$ varies according to conditions from 0.1 to 0.112.

In applying either of these formulas, experienced judgment is necessary to determine the proper coefficients to use; hence the results are likely to be misleading. Probably the practical man would get more accurate results from Thompson's empirical rule than from the more complex formula of Bodmer.

The investigation of condensation loss has gone on from the time of Hirn, and still we have no satisfactory formula for all cases. Those we have are based on variable, indefinite coefficients which are difficult to determine, and none take into account the ratio of compression which must certainly have a large influence. As to experimental data, the mere quotation of results which have been recorded with a given engine is futile, as the data could not be used in even an approximation with other than the original variables, a change in any one of which would destroy accuracy.

I have found food for thought in "Steam Using," by Prof. Smith, of Washington University, St. Louis, Mo., which quotes Hirn's experiments extensively, and also "The Steam Engine," by Prof. John Perry.

By applying Bodmer's formula with Mr. Abbott's engines it may be deduced that if clearance areas and ratio of expansion are identical in the two engines, the loss will also be identical, as in the 24x24-in. engine the surface exposed is relatively twice as large as in the 24x48-in., while the interval of time is only half as great. From a practical standpoint the short-stroke engine must have more compression and high compression produces high temperatures, so the range in the engine would probably be increased by high compression, which would increase its loss by condensation. This is probably the reason why long-stroke, low-speed engines nearly always excel short-stroke, high-speed engines in economy.

CHARLES F. PRESCOTT.

Los Angeles, Calif.

## Turbine Repair

Under this heading in the June 11 issue, Mr. Martin mentions that 0.009 in. clearance between the bushing and valve spindle of the pilot valve was a contributing cause of the turbine overspeeding on light loads. This amount is not excessive. Other turbine makers use no needle valve at all, as the steam for lifting the pilot valve is taken through the clearance between the bushing and spindle. Unless the opening is nearly as large as the relief ports around the pilot valve, I do not believe excessive overspeeding would result. Under this condition the valve would open violently and close slowly, which would cause racing on light loads. The spring might also be broken, if subject to such action.

It seems that Mr. Martin's turbine has been run without sufficient lubrication for the valve piston which has become dry, and, of course, would not work freely. When he injected oil through the needle valve, the piston was lubricated and the trouble vanished.

E. A. HAWLEY.

Des Moines, Iowa.

# Inquiries of General Interest

**Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it**

## *Speed of Belt over Belt*

When a belt wrapped around a driving pulley in the usual manner, carries another driving belt which is wrapped over the outside of the first belt, then in figuring the size or speed of a pulley driven by the outer belt, how much larger, if any, should the pulley be considered than its actual diameter?

H. R. C.

The pulley should be considered as increased in diameter by twice the thickness of the belt which is next to the pulley. For ordinary single leather belts allow for $\frac{3}{16}$ in. thickness, and for ordinary double belts allow for $\frac{3}{8}$ in. thickness of belt.

## *Object of Steam Pipe Anchorages*

Is it not practicable to construct anchorages for steam pipes strong enough to prevent expansion and contraction from changes of temperature?

D. W.

The force exerted by iron when restrained from freely expanding or contracting from change of temperature is so great that it is not practicable to provide for changes in dimension due to changes in temperature of piping, except by establishing stationary points or anchorages to or from which other points will move by action of contraction or expansion, and thus control the direction of motion without resisting the natural amount of motion. When anchorages are properly employed, the strength required of them is only sufficient to resist the reaction of forces which are opposed to the free movement of the pipe in assuming a natural position with respect to the part held stationary at the anchorage.

## *Locating the Vacuum Line on a Diagram*

Why is the vacuum line at sea level generally drawn on an indicator diagram below the atmospheric line at a distance in inches equal to 14.7 divided by the scale of the spring with which the diagram is made?

L. B.

Under average barometric conditions the pressure of the atmosphere at sea level is 14.7 lb. per sq.in. above zero, and as atmospheric pressure locates the atmospheric line at a height, which by the scale of the spring represents 14.7 lb. above zero, the zero or perfect vacuum line is to be drawn below the atmospheric line at a distance which represents 14.7 lb. per sq.in. less than atmospheric pressure.

As pounds pressure per square inch which any point on a diagram may be above or below atmospheric pressure is found in inches which the point is from that line, then, to find the distance in inches on the diagram which the vacuum line should come below the atmospheric line, we must reverse the operation; i.e., divide 14.7 lb. by the scale of the spring.

## *Back Water from Open Heater*

In using an open feed-water heater with a back-pressure valve on the exhaust for steam heating, is a partial vacuum on the heating system likely to cause flooding of the engine cylinder; and is there not danger of water getting to the cylinder from leakage of the water-supply valve to the heater when the plant is shut down?

G. W.

Standard types of open heaters usually have float valves for regulating the supply of water so as to maintain the proper level of water carried, and glass gages for observing whether the water stands at a safe level; so that with ordinary attention there should be little danger of flooding from the heater during the time the engine is running. When the water-supply valves are leaky, there may be danger of water filling the heater so full as to back over to the engine when a plant is shut down, and to guard against such an occurrence a stop valve should be in the exhaust pipe between the engine and the heater for cutting off the connection between them when the engine is shut down for any length of time.

## *Resistance of Round vs. Square Wire*

Each of two wires contains 100 cu.in. of annealed copper, one being round, 0.5 in. diameter, and the other a square wire 0.5 in. on a side. Which wire has the greater resistance? What is the length and resistance of each wire?

E. L.

The resistance may be found from the formula

$$R = p \times \frac{l}{A}$$

cross-sectional area in square inches, and $p$ the specific resistance of an inch cube of the conductor.

For annealed copper wire the specific resistance in microhms per inch cube at 0 deg. C. is 0.6276. Hence for the square wire with cross-sectional area of $0.5 \times 0.5 = 0.25$ sq.in., the length would be 400 in.; the resistance would be

$$R = \frac{0.6276 \times 400}{0.25};$$

$$= 1004.16 \text{ microhms};$$

$$= 0.001004 \text{ ohms}.$$

For the circular wire the cross-sectional area being $0.5 \times 0.5 \times 0.7854 = 0.19635$ sq.in., the length will be 509.29 in. and the resistance would be

$$R = \frac{0.6276 \times 509.29}{0.19635} = 1627.85 \textit{ microhms, approx.}$$

$$= 0.001628 \text{ ohms}.$$

## *Length of Eccentric Rod*

What is considered to be the length of an eccentric rod of an engine and how is the proper length determined?

L. G.

The length of the rod is the distance from the center of the eccentric to the center of the pin which connects the end of the rod to the rocker-arm or other part of the valve motion. Set the rocker-arm, or other part to which the pin is connected, in such a position that it will be in the middle of its travel. For setting the length of the eccentric rod of a Corliss engine, the rocker should stand plumb when connected to the wristplate in its central position. If the engine has no wristplate, then with a tram scribe a mark from the pin on the engine frame or other stationary object. Next obtain the extreme forward and backward travel of the eccentric by loosening the eccentric and turning it around the shaft, or by turning the engine over, and with the same length of tram as used before, lay off the extreme distances the pin has traveled in each direction; or if a Corliss engine, mark the extreme points of travel made by the center mark of the wristplate. Find the central point between the two positions thus laid off and the distance it is from the central mark will indicate the amount the rod should be lengthened or shortened to bring the pin to the correct position for mid-throw of the eccentric and be the proper length for the rod.

# Study Questions

**This Week's Questions**
**Last Week's Answers**

(66) When one of two engines in a power plant runs 20 per cent. underload and the other 20 per cent. overload their outputs are equal, but the combined output is 10 hp. less than the combined rated capacity. Under these conditions, what power does each engine develop and what are their rated capacities?

(67) The primary of a transformer contains 100 turns. How many turns must the secondary contain to increase the voltage from 110 to 2000?

(68) Each day for four days an engineer gives out 1 lb. of waste to one oiler, sends half the remaining stock to the boiler room and gives another pound to another oiler. He then has only 1 lb. left. How much had he in the beginning?

(69) How many gallons of water per minute will be raised to a height of 60 ft. by a hydraulic ram of 60 per cent. efficiency supplied with 100 cu.ft. of water per second under a head of 10 ft.?

(70) What is the absolute pressure in pounds per square inch corresponding to a manometer reading of 20 in. of water when the barometer reading is 29 in. of mercury?

*Answers to the above will appear in the next issue. Answers to last week's questions follow:*

(61) The heat equivalent of 1 hp.-hr. is 2545 B.t.u., and 1 hp. = 0.746 kw.; hence 1 kw. per minute would be equal to

$$\frac{2545}{60 \times 0.746} = 56.86 \; B.t.u.$$

The energy lost in the form of heat by the current passing through the coil is expressed by the formula

$$W = I^2R$$

where $W$ is watts, $I$ the current in amperes and $R$ the resistance in ohms.

Substituting

$$W = (22)^2 \times 5 = 2420 \; watts \; or \; 2.42 \; kw.$$

Since 1 kw. per minute is equivalent to 56.86 B.t.u., 2.42 kw. for 15 min. would be equivalent to

$$56.86 \times 2.42 \times 15 = 2064.02 \; B.t.u.$$

As 1 cu.ft. of water weighs 62.5 lb. and it takes 1 B.t.u. to raise 1 lb. of water 1 deg. F. at the temperature of the water, 2064.02 B.t.u. would raise 1 cu.ft.

$$\frac{2064.02}{62.5} = 33 \; deg. \; F.$$

(62) The total heat of combustion of a ton of the coal is

$$14{,}000 \times 2000 = 28{,}000{,}000 \; B.t.u.$$

Since 970.4 B.t.u. evaporates 1 lb. of water from and at 212 deg. F., the ton of coal will evaporate

$$\frac{28{,}000{,}000}{970.4} = 28{,}854 \; lb.$$

(63) The force actually exerted by the belt is the difference between the tensions on the two sides, or

$$600 - 300 = 300 \; lb.$$

and the distance through which it acts in 1 min. is the belt speed. This equals the circumference of the pulley multiplied by the revolutions per minute.

$$14 \times 3.1416 \times 180 = 7916.832 \; in.$$

or

$$\frac{7916.832}{12} = 659.736 \; ft.$$

The work done is therefore

$$300 \times 659.736 = 197{,}920.8 \; ft.\text{-}lb.$$

and the horsepower

$$\frac{197{,}920.8}{33{,}000} = 5.9976 \; hp.$$

(64) For simplicity the whole weight is considered as concentrated at the end of the mean radius. The mean velocity per second is

$$V = \frac{2 \times 3.1416 \times 5 \; ft. \times 70 \; r.p.m.}{60 \; sec.}$$

$$= 36.652 \; ft.$$

The energy is

$$E = \frac{w v^2}{2g}$$

where

$w$ = Weight;
$v$ = Velocity;
$2g$ = Twice the acceleration of gravity = 64.4.

Then

$$E = \frac{7000 \times 36.652^2}{64.4} = 146{,}018.38 \; ft.\text{-}lb.$$

(65) To save remeasuring all of the indicator cards, the horsepower can be corrected by the ratios of the lengths. It is known the mean effective pressure obtained, and therefore the horsepower, must be too large, so the correction is

$$\frac{Hp.}{3\frac{1}{2}} \times 3 \; or \; \frac{435}{3.5} \times 3 = 372.857 \; i.hp.$$

This is explained in the answer to the inquiry, "Using Averaging Planimeter" in the July 30 issue, page 167.

A new apparatus for separating oil from exhaust steam, described by the *Engineer*, of London, consists of a series of slightly curved plates standing vertically in a box through which the steam is passed. Each plate has four or five vertical baffles on the concave side. The plates are stacked close together, and their slight curvature, while sufficient to intercept oil and water in the steam, is not sufficient to cause much back pressure. An efficiency test is reported in which only $\frac{3}{10}$ of 1 per cent. of the original oil fed in passed through the separator to the condenser.

# Derivation of the Massachusetts Formula for Areas of Circular Segments

By W. C. Marshall*

Some years ago, S. F. Jeter, then engineer for the Bigelow Co., sent the writer, through one of the students in the Sheffield Scientific School, a slip of paper on which was the Massachusetts formula for the area of a circular segment, viz.:

$$A = \tfrac{4}{3} H^2 \sqrt{\frac{2R}{H} - 0.608}$$

asking how it was obtained.

After some calculating the writer reached the conclusion that it was obtained from the rule for the area of a parabolic segment, which is

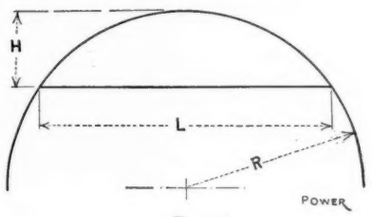



FIG. 1

$$A = \tfrac{2}{3} H L$$

where

$L$ = Length of chord;
$H$ = Height of segment;

as in Fig. 1.

The parabolic rule does not take into account the radius of the circle when it is used on a circular segment. Suppose it is reduced to an expression which brings in the radius. If $L$ = one-half the chord length, then

$$A = \tfrac{4}{3} HL$$

The value of $L$ may be obtained from the right triangle *abo*, Fig. 2, from

$$R^2 = L^2 + (R - H)^2$$

or

$$L^2 = R^2 - (R - H)^2$$

from which

$$L = H \sqrt{\frac{2R}{H} - 1}$$

Substituting this value of $L$ in the expression

$$A = \tfrac{4}{3} HL$$

gives

$$A = \tfrac{4}{3} H^2 \sqrt{\frac{2R}{H} - 1}$$

Comparing this formula with the Massachusetts formula the only difference is found to be in the constant under the radical, which is 1 instead of 0.608. The constant 0.608 was probably obtained as follows:

*Assistant Professor of Machine Design Sheffield Scientific School, Yale University.

Take a semicircle, apply the parabolic rule for its area and

$$A = 1.33\ R^2$$

The area of a semicircle is

$$A = \frac{\pi R^2}{2}$$

or

$$A \times \frac{3.1416\ R^2}{2} = 1.57\ R^2$$

showing the parabolic rule to be in error by giving an area too small by the amount 1.33: 1.57, or if the areas are compared by the formulas

$$A = 1.33\ R^2 \sqrt{\ 1} \text{ (Parabolic)} \quad (1)$$

$$A = 1.57\ R^2 \text{ (Actual)} \quad (2)$$

Equation (2) can be reduced to the *form* of (1) by dividing 1.57 by 1.33 and writing under the square-root sign the square of the quotient 1.178, or

$$\left[\frac{3.1416}{2} \div \tfrac{4}{3} = \frac{3.1416}{2} \times \tfrac{3}{4} = \frac{1.57+}{1.33+} = 1.178\right]$$

$$A = 1.33 \times 1.178\ R^2 = 1.33\ R^2 \sqrt{\ 1.387} \quad (3)$$

But the number under the radical in (1) is $\sqrt{\ .2 - 1}$ while in (3) it is $\sqrt{\ 2 - 0.61}$. Thus it is evident that the Massachusetts rule is the parabolic formula modified by changing the constant under the radical from 1 to 0.608 so as to give the true area for the largest possible segment, viz., a semicircle.

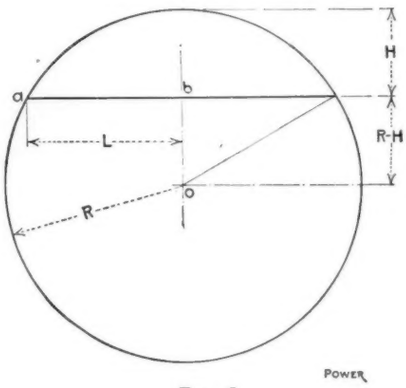



FIG. 2

The parabolic rule is less in error as the segment is smaller, for example: Take the area of the segment for a sector of ¼ the area of a circle. The area by the parabolic rule is 0.276, while the Massachusetts rule gives 0.285, which is correct. The error by the parabolic rule in this case is — 3 per cent. of the total area considered.

In the great Krupp plant in Germany, 89,430 hp. is the total steam power, and 2690 electric motors have an output of 50,491 hp. There are 68,500 employees.

It is predicted that concrete radiators for steam or hot-water heating will soon be made practical. They are a success in Germany, it is said.

## The Manufacturers' Standard of Flanged Fittings

On other pages of this issue we have compared the two standard schedules for flanged fittings evolved by the engineering societies and the manufacturers. The differences between the two are clearly shown, and in this connection it may be interesting to learn why the manufacturers refused to conform with the societies' standard. Following is a statement sent out by the secretary of the manufacturers' committee.

"Without question, a universal standard is necessary and highly desirable; but any standard to be commercially successful must be designed to bring about the desired uniformity with the minimum inconvenience and expense, both to consumers and to manufacturers.

"Before the 1912 U. S. Standard was adopted there existed among the majority of manufacturers, in the United States, a generally accepted standard for flanged fittings and flanges. Most of the large flanged fittings manufacturers in this country had the entire pattern equipment necessary to build this line of goods and many had list prices and dimensions published for general distribution to the trade; in fact, these dimensions have been published in Kent's 'Pocket Book,' eighth edition.

"After having thoroughly investigated the conditions which induced the two societies to adopt the 1912 U. S. Standard, the manufacturers of flanged fittings and flanges throughout the country realized that certain changes in the generally accepted standard were desirable. The tendency of modern engineers to use higher steam-working pressures than was their practice some years ago has demonstrated the necessity for an increase in the strength of extra-heavy end flange bolting in the larger sizes. In drafting up the 1912 U. S. Standard, the American Society of Mechanical Engineers provided an increase in the bolting of large extra-heavy flanges; but this increase seems out of proportion with the necessities of steam-engineering practice. The manufacturers have, therefore, adopted a happy medium and have increased the bolting on these extra-heavy flanges, but have been more practical in this increase than the 1912 U S. Standard.

"So far as the strength of the fittings themselves, not considering the bolting, is concerned, the 1912 U. S. Standard has no advantage whatever over the Manufacturers' Standard.

"While the 1912 U. S. Standard provides longer center-to-face dimensions on many straight sizes, this change was not brought about by any attempt to better the fittings, but was an average of several published sets of dimensions which in many cases were out of date and did not correspond with the goods being made by the respective manufacturers.

"This change alone would require a large expenditure for patterns and shop equipment, without a commensurate advantage to the consumers.

"The increase in bolting in the Manufacturers' Standard compares very favorably with the table of dimensions on 'A Standard for High-Pressure Steam Lines,' issued by the Society of German Engineers in 1900; also, the report issued by the Engineering Standard Committee on British Standard Tables of Pipe Flanges.

"While the 1912 U. S. Standard provides much heavier bolting on the very large extra-heavy flanges, this excessive increase does not seem necessary, as the Manufacturers' Standard bolting is just as strong on the sizes, which include the great majority of pipe lines; namely, the 10-in. size and smaller.

"The Manufacturers' Standard makes no change in the center-to-face dimensions from the present standard, which avoids mistakes when the consumer makes changes or additions to his plant. It is also economical for the manufacturer, as he can use his present equipment, which, to a large extent, is made up in a very substantial fashion and cannot be changed without involving a high expenditure.

"Furthermore, the 1912 U. S. Standard would necessitate carrying two stocks of fittings for several years, or until the new standard is fully in effect. The Manufacturers' Standard will not cause this difficulty.

"The Manufacturers' Standard for reducing fittings is more in proportion with the straight sizes. The 1912 U. S. Standard has the same dimensions on the run of reducing fittings as the straight sizes, regardless of reduction. Consequently the proportion of a reducing fitting will not be in harmony with those of a straight fitting.

"It would seem apparent that the excessively large bolt diameters specified in the 1912 U. S. Standard for the larger flanges are much too cumbersome and inconvenient from the steamfitters' standpoint, while the bolt sizes specified in the Manufacturers' Standard are more in accordance with general practice. In a standard of this nature convenience as well as strength should be a governing factor.

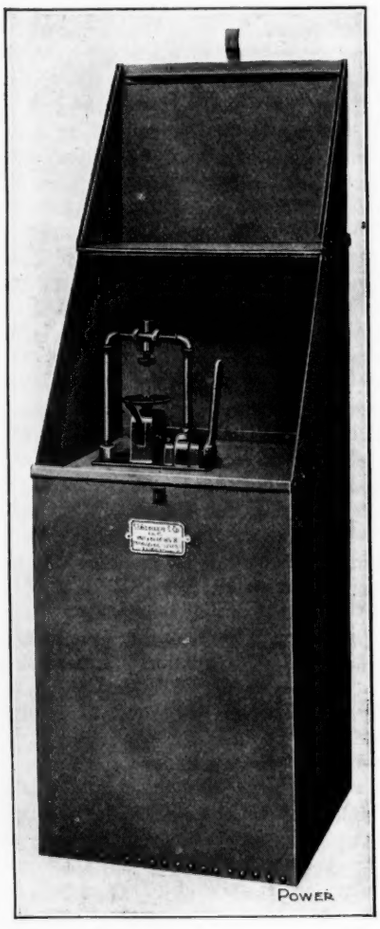
"While the manufacturers may be accused of not taking an interest in the 1912 U. S. Standard from its inception, their recent action in adopting a universal standard which is fully in accord with the requirements of the engineering trade, will demonstrate that they are acting with absolute sincerity and with a fair-minded attitude toward all interests involved."

In France a porous glass is being used for ventilation, having holes small enough to keep out drafts and dust.

## Bowser Non-Overflow Oil Can Filler

S. F. Bowser & Co., Fort Wayne, Ind., are placing on the market an outfit designed for storing lubricating oil, and having a quick-acting pump which is used in filling hand oilers. The pump has an adjustable table operated with a lever. The hand oiler is placed on this table and the lever is pushed down, throwing the oiler into the proper position and fitting the nozzle into the opening of the hand oiler.

The height of the table can be adjusted instantly to accommodate it to oilers of different heights. When the oiler is in the filling position, the nozzle of the pump fits into the oil can, making an oil-tight connection. A few quick strokes of the pump fills the can. Any amount over the capacity of the oiler is returned to the storage tank through an overflow line, which prevents spilling the oil over the can.




A Non-overflowing Filler for Hand Oilers

The cabinet is provided with a metal hood. The outfit is fireproof and is made of sheet steel finished in bronze green.

During 1911, the exports of British coal amounted to 64,599,266 long tons.

## The Two Schedules of Flanged Fittings

Two standards for flanges and flanged fittings are now before the public. The "U. S. 1912 Standard," evolved by the American Society of Mechanical Engineers and the National Association of Steam and Hot Water Fitters, was published in the Mar. 19, 1912, issue of Power. In the accompanying table the light figures reproduce this schedule and the heavy-face type opposite gives the variations found in the "Manufacturers 1912 Schedule," which was adopted July 10 and is to take effect Oct. 1.

For the standard fittings it may be noticed that the greatest variation occurs in the dimensions for the long turn elbow and for tees, crosses and laterals with reducing outlets.

For tees and crosses with reducing outlets from 9 in. up, the societies vary the center to face of outlet dimension from the standard only when the branch outlet is 8 in. in diameter and under. From 4½ in. to 8 in. in the laterals a variation is made for all reductions below 4 in., and from 9 in. up, a reduction to 8 in. or under calls for a different length from center to face of outlet. The face-to-face dimensions of the fittings are maintained, but not so in the manufacturers' schedule, which is given *under* the societies' figures in these cases in the heavy-face type.

No dimensions for laterals 1, 1¼, 1½, 26, 28 and 30 in. in diameter are given by the manufacturers, nor do they include any figures for Y-fittings, as these are regarded as special and are made to suit conditions. On the other hand, they give a set of dimensions for reducers, while the societies rule that the face-to-face dimension shall be equal to the diameter of the larger flange. The manufacturers' figures on reducers are given in the table. For the societies' tabulation on Y-fittings reference may be made to the Mar. 19 issue.

More variation occurs among the extra-heavy flanged fittings and particularly among the larger sizes and the fittings reducing on the branch. The same plan of tabulation was followed and the comparison should be interesting, as it shows at a glance the difference between the two standards.

### Explanatory Notes

Some differences also occur in the explanatory notes accompanying the schedules. These will be evident from the following, in which the societies' notes appear in Roman type, and the manufacturers' variations, wherever they occur, follow in *italics*.

1. Standard or extra-heavy reducing elbows carry the same dimensions center to face as the regular elbows of the largest straight size.

2. Standard or extra-heavy tees, crosses and laterals, reducing on run, carry the same dimensions face to face as the largest straight size.

3. If flanged fittings for lower working pressures than 125 lb. are made, they shall conform in all dimensions, except in thickness of shell, to this standard, and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be of standard dimensions.

4. Where long-turn fittings are specified, it has reference only to elbows, which are made in two center to face dimensions, to be known as "elbows" and "long-turn" elbows, the latter being used only when so specified.

5. All standard-weight fittings must be guaranteed for 125 lb. and extra-heavy fittings for 250 lb. working pressure, and each fitting must have some mark cast on it indicating the maker and the guaranteed working steam pressure.

6. All extra-heavy fittings and flanges to have a raised surface $\frac{1}{16}$ in. high inside of the bolt holes for the gasket.

Standard-weight fittings and flanges to be plain faced.

Bolts to be ⅛ in. smaller in diameter than the bolt holes.

Bolt holes should straddle the center lines.

7. Size of all fittings scheduled indicates the inside diameter of the ports. For the outside diameter of pipe use the corresponding size of the inside diameter fittings.

7. *Size of all fittings scheduled indicates inside diameter of ports except for extra-heavy fittings 14 in. and larger when the port diameter is ¾ in. smaller than nominal size.*

8. The face-to-face dimension of a reducer, either straight or eccentric, shall be equal to the diameter of the larger flange.

8. *The face-to-face dimension of reducers, either straight or eccentric, for all pressures, shall be the same face to face as given in the table of dimensions.*

9. Square head bolts with hexagonal nuts are recommended.

10. Twin ells, double branch ells, side outlet ells, side outlet tees and four-way tees, whether straight sizes or reducing, carry the same dimensions center to face and face to face as the regular ells and tees.

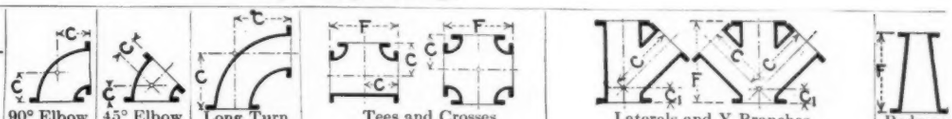
10. *Twin elbows, whether straight or reducing, carry same dimensions, center to face and face to face, as regular straight size ells and tees.*

*Side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center to face and face to face as regular tee having the same reductions.*

11. Bull-head tees or tees increasing on outlet will have the same center to face and face to face dimensions as a straight fitting of the size of the outlet.

## SCHEDULE OF FLANGES AND FLANGED FITTINGS.

Standard Weight Flanged Fittings.




| Size, Inches | All Fittings and Flanges: Diameter of Flange | Thickness of Flange | Number of Bolts | Diameter of Bolt Circle | Diameter of Bolt Holes | Size of Bolts | 90° Elbow: Center to Face "C" | 45° Elbow: Center to Face "C" | Long Turn Elbow: Center to Face "C" | Tees and Crosses: Center to Face "C" | Tees and Crosses: Center to Face Outlet 8" and Under | Tees and Crosses: Face to Face "F" | Laterals and Y Branches: Center to Face "C" | Laterals and Y Branches: Center to Face on Short End "$C_1$" | Laterals and Y Branches: Center to Face of Outlets Smaller in Dia. | Reducer: Face to Face "F" | Reducer: Face to Face "F" |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 4 | 7/16 | 4 | 3 | 5/8 | 7/16 | 3½ | 2 | 1⅝ 5½ | 5 | 3½ | | 7 | 6 | 2 | | 8 | |
| 1¼ | 4½ | ½ | 4 | 3⅜ | ⅝ | ½ | 3½ | 2 | 6 | 5½ | 3½ | | 7½ | 6½ | 2½ | — | 8½ | |
| 1½ | 5 | ½ | 4 | 3¾ | ⅝ | ½ | 4 | 2½ | 6½ | 6 | 4 | | 8 | 6½ | 2½ | — | 9½ | |
| 2 | 6 | ½ | 4 | 4¾ | ⅝ | ½ | 4½ | 2½ | 7 | 6½ | 4½ | | 9 | 8 | 2½ | — | 10½ | |
| 2½ | 7 | ½ | 4 | 5½ | ⅝ | ½ | 5 | 2½ 3 | 7½ | 7 | 5 | | 10 | 9½ | 2½ | — | 12 | |
| 3 | 7½ | ½ | 4 | 6 | ⅝ | ½ | 5½ | 3 | 8 | 7½ | 5½ | | 11 | 10 | 3 | | 13 | 6 |
| 3½ | 8½ | ½ | 4 | 7 | ⅝ | ½ | 6 | 3½ | 9 | 8½ | 6 | | 12 | 11½ | 3 | | 14½ | 6½ |
| 4 | 9 | ½ | 8 | 7½ | ⅝ | ½ | 6½ | 4 | 10 | 9 | 6½ | | 13 | 12 | 3 | | 15 | 7 |
| | | | | | | | Outlet 2½ in. and under | | | | | | | | 2 | 11 | 13 | |
| 4½ | 9½ | ½ | 8 | 7½ | ⅞ | ¾ | 7 | 4½ 4 | 11 | 9½ | 7 | | 14 | 12½ | 3 | 11½* | 15½ | 7½ |
| | | | | | | | Outlet 2½ in. and under | | | | | | | | 2 | 11 | 13 | |
| 5 | 10 | ½ | 8 | 8½ | ⅞ | ¾ | 7½ | 4½ | 12 | 10½ | 7½ | | 15 | 13½ | 3½ | 12½* | 17 | 8 |
| | | | | | | | Outlet 3 in. and under | | | | | | | | 2 | 12 | 14 | |
| 6 | 11 | 1 | 8 | 9½ | ⅞ | ¾ | 8 | 5 | 13 | 11½ | 8 | | 16 | 14½ | 3½ | 13* | 18 | 9 |
| | | | | | | | Outlet 3 in. and under | | | | | | | | 1½ | 13½ | 15 | |
| 7 | 12½ | 1 1/16 | 8 | 10¾ | ⅞ | ¾ | 8½ | 5½ | 14½ | 12½ | 8½ | | 17 | 16½ | 4 | 13½* | 20½ | 10 |
| | | | | | | | Outlet 3½ in. and under | | | | | | | | 1½ | 15 | 16 | |
| 8 | 13½ | 1⅛ | 8 | 11¾ | ⅞ | ¾ | 9 | 6 5½ | 16 | 14 | 9 | | 18 | 17½ | 4½ | 14½* | 22 | 11 |
| | | | | | | | Outlet 4 in. and under | | | | | | | | 1½ | 15½ | 16 | |
| 9 | 15 | 1⅛ | 12 | 13¼ | ⅞ | ¾ | 10 | 6½ 6 | 18 | 15¼ | 10 | 9 | 20 | 19½ | 4½ | 17½† | 24 | 11½ |
| | | | | | | | Outlet 4½ in. and under | | | | | | | | 1½ | 16½ | 17 | |
| 10 | 16 | 1 3/16 | 12 | 14¼ | 1 | ⅞ | 11 | 7 6½ | 20 | 16½ | 11 | 10 | 22 | 20½ | 5 | 18½† | 25½ | 12 |
| | | | | | | | Outlet 5 in. and under | | | | | | | | 1 | 18 | 18 | |
| | | | | | | | Outlet 6 in. and under | | | | | 9½ | 18 | | | | | |
| 12 | 19 | 1¼ | 12 | 17 | 1 | ⅞ | 12 | 7½ | 22 | 19 | 12 | 11½ | 24 | 24½ | 5½ | 20† | 30 | 14 |
| | | | | | | | Outlet 6 in. and under | | | | | | | | 1 | 20½ | 20 | |
| | | | | | | | Outlet 8 in. and under | | | | | 11 | 20 | | | | | |
| 14 | 21 | 1⅜ | 14 | 18¾ | 1⅛ | 1 | 14 | 7½ | 24 | 21½ | 14 | 12½ | 28 | 27 | 6 | 21½† | 33 | 16 |
| | | | | | | | Outlet 7 in. and under | | | | | | | | 1 | 23 | 22 | |
| | | | | | | | Outlet 9 in. and under | | | | | 13 | 22 | | | | | |
| 15 | 22½ | 1⅜ | 16 | 20 | 1⅛ | 1 | 14½ | 8 | 26 | 22¾ | 14½ | 13 | 29 | 28½ | 6 | 22⅛† | 34½ | 17 |
| | | | | | | | Outlet 7 in. and under | | | | | | | | 1 | 24 | 23 | |
| | | | | | | | Outlet 9 in. and under | | | | | 13½ | 23 | | | | | |
| 16 | 23½ | 1 5/16 | 16 | 21¼ | 1⅛ | 1 | 15 | 8 | 28 | 24 | 15 | 13½ | 30 | 30 | 6½ | 22⅞† | 36½ | 18 |
| | | | | | | | Outlet 8 in. and under | | | | | | | | 1 | 25½ | 24 | |
| | | | | | | | Outlet 10 in. and under | | | | | 14 | 24 | | | | | |
| 18 | 25 | 1 7/16 | 16 | 22¾ | 1¼ | 1⅛ | 16½ | 8½ | 30 | 26½ | 16½ | 14½ | 33 | 32 | 7 | 24½† | 39 | 19 |
| | | | | | | | Outlet 9 in. and under | | | | | | | | 1 | 27½ | 26 | |
| | | | | | | | Outlet 12 in. and under | | | | | 15½ | 26 | | | | | |
| 20 | 27½ | 1 11/16 | 20 | 25 | 1¼ | 1⅛ | 18 | 9½ | 32 | 29 | 18 | 15½ | 36 | 35 | 8 | 25½† | 43 | 20 |
| | | | | | | | Outlet 10 in. and under | | | | | | | | 1 | 29½ | 28 | |
| | | | | | | | Outlet 14 in. and under | | | | | 17 | 28 | | | | | |
| 22 | 29½ | 1 13/16 | 20 | 27¼ | 1¼ | 1⅛ | 20 | 10 | 34 | 31½ | 20 | 16½ | 40 | 37½ | 8½ | 27½† | 46 | 22 |
| | | | | | | | Outlet 10 in. and under | | | | | | | | ½ | 31½ | 29 | |
| | | | | | | | Outlet 15 in. and under | | | | | 18 | 28 | | | | | |
| 24 | 32 | 1⅞ | 20 | 29½ | 1¼ | 1⅛ | 22 | 11 | 36 | 34 | 22 | 17½ | 44 | 40½ | 9 | 28½† | 49½ | 24 |
| | | | | | | | Outlet 12 in. and under | | | | | | | | ½ | 34½ | 32 | |
| | | | | | | | Outlet 16 in. and under | | | | | 19 | 30 | | | | | |
| 26 | 34½ | 2 | 24 | 31¾ | 1⅜ | 1¼ | 23 | 13 | 39 | 36½ | 23 | 19½ | 46 | 43½ | 9½ | 30¾† | 53 | 26 |
| | | | | | | | Outlet 18 in. and under | | | | | 20 | 32 | | | | | |
| 28 | 36½ | 2 1/16 | 28 | 34 | 1⅜ | 1¼ | 24 | 14 | 42 | 39 | 24 | 20½ | 48 | 46½ | 10½ | 33† | 57 | 28 |
| | | | | | | | Outlet 18 in. and under | | | | | 21 | 32 | | | | | |
| 30 | 38¾ | 2⅛ | 28 | 36 | 1½ | 1⅜ | 25 | 15 | 45 | 42 | 25 | 21½ | 50 | 49½ | 11 | 35½† | 60½ | 30 |
| | | | | | | | Outlet 20 in. and under | | | | | 23 | 36 | | | | | |

### Extra Heavy Flanged Fittings

| | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 4½ | 11/16 | 4 | 3½ | ⅝ | ½ | 4½ 4 | 2½ 2 | 5½ 5 | 4½ 4 | | 8½ 8 | 7½ | 2½ | — | 9½ | | |
| 1¼ | 5 | ¾ | 4 | 3¾ | ⅝ | ½ | 4½ | 2½ | 6 5½ | 4½ | | 8½ | 7½ | 2½ | — | 9½ | | |
| 1½ | 6 | ¾ | 4 | 4½ | ⅝ | ½ | 4½ | 2½ | 6½ 6 | 4½ | | 9 | 8½ | 2½ | — | 11 | | |
| 2 | 6½ | ¾ | 4 | 5 | ⅝ | ½ | 5 | 3 | 7 6½ | 5 | | 10 | 9½ 9 | 2½ | — | 12 | 11½ | |
| 2½ | 7½ | 1 | 4 | 5½ | ⅝ | ½ | 5½ | 3½ 3½ | 7½ 7 | 5½ | | 11 | 10½ | 2½ | — | 13 | | |
| 3 | 8½ | 1⅛ | 8 | 6½ | ⅝ | ½ | 6 | 3½ | 8 7½ | 6 | | 12 | 11½ 11 | 3 | — | 14½ | 14 | 6 |
| 3½ | 9 | 1 1/16 | 8 | 7¼ | ⅝ | ½ | 6½ | 4 | 9 8½ | 6½ | | 13 | 12½ | 3 | — | 15½ | | 6½ |
| 4 | 10 | 1¼ | 8 | 7⅞ | ⅝ | ½ | 7 | 4½ 4½ | 10 9 | 7 | | 14 | 13½ | 3 | — | 16½ | | 7 |
| | | | | | | | Outlet 2½ in. and under | | | | | | | 2 | 13 | 15 | | |
| 4½ | 10½ | 1 1/16 | 8 | 8½ | ⅞ | ¾ | 7½ | 4½ | 11 9½ | 7½ | | 15 | 14½ | 3½ | 13½* | 18 | | 7½ |
| | | | | | | | Outlet 2½ in. and under | | | | | | | 2½ | 13½ | 15 | | |
| 5 | 11 | 1⅛ | 8 | 9¼ | ⅞ | ¾ | 8 | 5 | 12 10¼ | 8 | | 16 | 16 15 | 3½ | 14* | 19½ | 18½ | 8 |
| | | | | | | | Outlet 3 in. and under | | | | | | | 2½ | 14½ | 16 | | |
| 6 | 12½ | 1 1/16 | 12 | 10¾ | ⅞ | ¾ | 9 8½ | 5½ | 13 11½ | 9 8½ | | 18 17 | 17½ | 4 | 14½* | 21½ | | 9 |
| | | | | | | | Outlet 3 in. and under | | | | | | | 2½ | 15½ | 17 | | |
| 7 | 14 | 1⅛ | 12 | 11¾ | 1 | ⅞ | 9½ 9 | 5½ 6 | 14½ 12½ | 9½ 9 | | 19 18 | 19 18½ | 4½ | 15½* | 23½ | 23 | 10 |
| | | | | | | | Outlet 3½ in. and under | | | | | | | 2½ | 16½ | 18 | | |
| 8 | 15 | 1⅛ | 12 | 13 | 1 | ⅞ | 10½ 10 | 6 | 16 14 | 10½ 10 | | 21 20 | 20½ 20 | 5 | 16* | 25½ | 25 | 11 |
| | | | | | | | Outlet 4 in. and under | | | | | | | 2½ | 18½ | 20 | | |
| 9 | 16½ | 16¼ | 1¼ | 12 | 14 | 1⅛ | 1 11 10½ | 6½ | 18 15½ | 11 10½ | 10½ | 22 21 | 22¾ 21 | 5 | 20½† | 27½ | 26 | 11½ |
| | | | | | | | Outlet 4½ in. and under | | | | | | | 2½ | 19½ | 21 | | |
| 10 | 18½ | 17½ | 1⅜ | 16 | 15¾ | 15¼ | 1⅛ 1 12 11½ | 7 | 20 16½ | 12 11½ | 11 | 24 23 | 25¼ 23 | 5½ | 21½† | 30½ | 28½ | 12 |
| | | | | | | | Outlet 5 in. and under | | | | | | | 2½ | 21½ | 23 | | |
| | | | | | | | Outlet 6 in. and under | | | | | 11 | 18 | | | | | |
| 12 | 20½ | 20 | 2 | 16 | 17¾ | | 1¼ 1⅛ 1⅛ 1 13½ 13 | 8 | 22 19 | 13½ 13 | 12 | 27 26 | 27¾ 26½ | 6 | 23† | 33½ | 32½ | 14 |
| | | | | | | | Outlet 6 in. and under | | | | | | | 2½ | 24½ | 26 | | |
| | | | | | | | Outlet 8 in. and under | | | | | 12½ | 21 | | | | | |
| 14 | 23½ | 22½ | 2⅛ | 20 | 20¼ | 20 | 1⅜ 1⅛ 1¼ 1 15 14½ | 9½ 8 | 24 21½ | 15 14½ | 13 | 30 29 | 31¼ 30 | 6½ | 24½† | 37½ | 36½ | 16 |
| | | | | | | | Outlet 7 in. and under | | | | | | | 2½ | 27½ | 29 | | |
| | | | | | | | Outlet 9 in. and under | | | | | 13½ | 22 | | | | | |
| 15 | 25 | 23½ | 2 1/16 | 20 | 21½ | 21 | 1⅜ 1¼ 1¼ 1⅛ 15½ 15 | 10 8½ | 26 22¾ | 15½ 15 | 13¼ | 31 30 | 33¼ 31½ | 6½ | 25½† | 39½ | 38 | 17 |
| | | | | | | | Outlet 7 in. and under | | | | | | | 2½ | 28½ | 30 | | |
| | | | | | | | Outlet 9 in. and under | | | | | 13½ | 23 | | | | | |
| 16 | 26 | 25 | 2¼ | 20 | 22½ | | 1½ 1¼ 1⅜ 1⅛ 16½ 16 | 10½ 9 | 28 24 | 16½ 16 | 14 | 33 32 | 34½ 33 | 7½ | 25¾† | 42 | 40½ | 18 |
| | | | | | | | Outlet 8 in. and under | | | | | | | 3 | 30½ | 32 | | |
| | | | | | | | Outlet 10 in. and under | | | | | 15 | 24 | | | | | |
| 18 | 28½ | 27 | 2⅜ | 24 | 24¾ | 24½ | 1½ 1¼ 1⅜ 1⅛ 17½ 17 | 11 9½ | 30 26½ | 17½ 17 | 15 | 35½ 34 | 37¾ 36 | 8 | 27½† | 45½ | 44 | 19 |
| | | | | | | | Outlet 9 in. and under | | | | | | | 3 | 32½ | 34 | | |
| | | | | | | | Outlet 12 in. and under | | | | | 16½ | 27 | | | | | |
| 20 | 31 | 29½ | 2½ | 24 | 27 | 26¾ | 1⅜ 1⅜ 1½ 1¼ 19¼ 18½ | 11½ 10 | 32 29 | 19¼ 18½ | 16 | 38½ 37 | 40¾ 38 | 8½ | 28½† | 49½ | 46½ | 20 |
| | | | | | | | Outlet 10 in. and under | | | | | | | 3 | 36 | 37 | | |
| | | | | | | | Outlet 14 in. and under | | | | | 17½ | 30 | | | | | |
| 22 | 33 | 31½ | 2⅝ | 28 | 29¼ | 28¾ | 1⅜ 1⅜ 1½ 1¼ 20½ 20 | 12 10½ | 34 31½ | 20½ 20 | 17 | 41 40 | 43¾ 41½ | 9½ | 30† | 53½ | 51 | 22 |
| | | | | | | | Outlet 10 in. and under | | | | | | | 3 | 39 | 40 | | |
| | | | | | | | Outlet 15 in. and under | | | | | 18½ | 30 | | | | | |
| 24 | 36 | 34½ | 2¾ | 28 | 32 | 31½ | 1⅜ 1½ 1⅜ 1⅜ 22½ 22 | 13 11½ | 36 34 | 22½ 22 | 18 | 45 44 | 47½ 44½ | 10 | 31½† | 57½ | 54½ | 24 |
| | | | | | | | Outlet 12 in. and under | | | | | | | 3 | 43 | 44 | | |
| | | | | | | | Outlet 15 in. and under | | | | | 19½ | 30 | | | | | |

*Outlets 4 in. and under. †Outlets 8 in. and under.

12. Up to and including the 4-in. size, center to face and face to face dimensions of reducing fittings will be the same as that of a straight fitting of the larger opening.

12. *Tees and crosses 9 in. and down, reducing on the outlet or run and outlet, use the same dimensions as straight sizes of the larger port.*

*Sizes 10 in. and up, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions.*

*If the outlet is larger than that given in table, use dimensions of straight sizes.*

*Laterals 3½ in. and down, reducing on the branch, use the same dimensions as straight sizes of the larger port.*

*Sizes 4 in. and up, reducing on the branch, are made in two lengths, depending on the size of the branch as given in the table of dimensions.*

*If the outlet is larger than that given in the table, use dimensions of the straight sizes.*

*Y's are special and are made to suit conditions.*

*Double sweep tees are not made reducing on the run.*

*Steel flanges, fittings and valves are recommended for superheated steam.*

13. Pipe sizes 14 in. and over refer to outside diameter.

# Some New Condensing Apparatus

**Some of the more notable products in the line of condensing machinery built by the C. H. Wheeler Manufacturing Co., as seen on a recent visit to the works at North Philadelphia, Pa.**

The writer recently paid a very interesting visit to the works of the C. H. Wheeler Manufacturing Co., of Philadelphia, Penn., and saw many features in the design of condensing machinery which will be of interest to the readers of POWER.

The plant is located at North Philadelphia and occupies the site of the factory established by William Barr in the early nineties, and where many record-breaking pumping engines were built by him.

Control of this plant was taken over by C. H. Wheeler (whose name is so intimately associated with condensing machinery) about eight years ago and the works remodeled to engage in the manufacture of condensing apparatus, particularly the high efficiency designs necessary for turbine practice.

Fig. 1 shows an arrangement or tube layout for a turbine surface condenser, designed for the exhaust steam to enter on one side and to operate on the wet

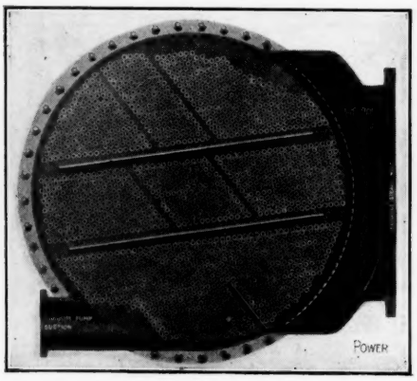



FIG. 1. DRAIN-CIRCULATION PLATES AND TUBE ARRANGEMENT IN A TURBINE CONDENSER

system, one pump removing the condensed steam, air and vapor. The drain circulation plates, shown together with the entering steam lanes provided by the omission of tubes at certain points, it is claimed, keep the entire tube surface active and enable the condensed steam to be removed at a temperature closely approximating that of the vacuum. This removes a serious objection many engineers have to a combined system of this type, viz., the chilling of the feed water, which was generally considered unavoidable in the operation of a strictly wet system.

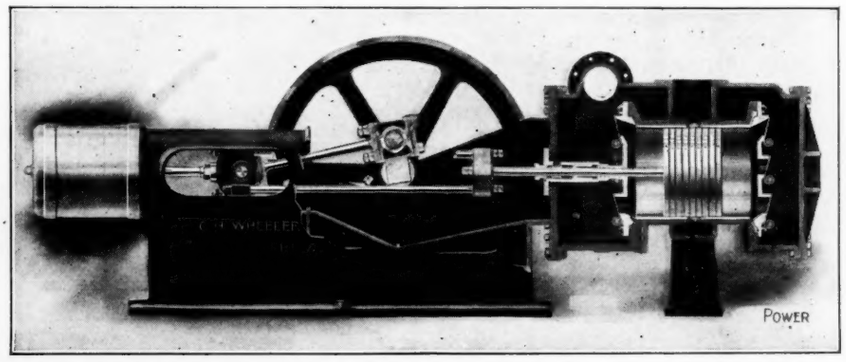



FIG. 2. BROKEN VIEW OF THE MULLAN SUCTION VALVELESS VACUUM PUMP

Inquiring as to the design of vacuum pump used in connection with such a condenser, the writer was shown a number of the Mullan pumps. Fig. 2 reveals the construction of the vacuum cylinder of these machines. The design is attractive in its extreme simplicity; the pump is of the double-acting horizontal type and all the working parts are bronze composition to withstand the action of the distilled water handled and avoid corrosion and deterioration, if shut down for any length of time. The pump is suction valveless, the place of these valves being taken by a series of ports cut in the cylinder liner, and the piston in its travel to and fro, alternately connects the two ends of the cylinder with the condenser, then discharges this vapor and water on the return stroke through the valves on the vertical valve decks at each end of the cylinder. This pump has previously been described in POWER, but the recent improvements noted were the increased port areas and the new type of coil flexible metallic valves, enabling the pump to operate at a higher speed than the earlier designs and increasing its volumetric and mechanical efficiencies.

For circulating service a regular centrifugal pump is used and a number of these machines were seen under construction, some engine, some motor and some steam-turbine driven. Fig. 3 is a view, partly in section, of such a pump. It is of the double-suction volute design

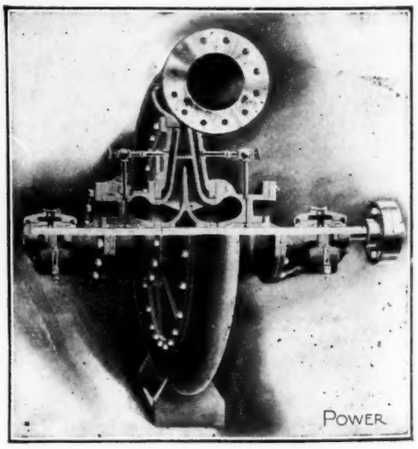



FIG. 3. CENTRIFUGAL CIRCULATING PUMP PARTLY IN SECTION

with outboard independent ring-oiling bearings.

One of the most interesting develop-

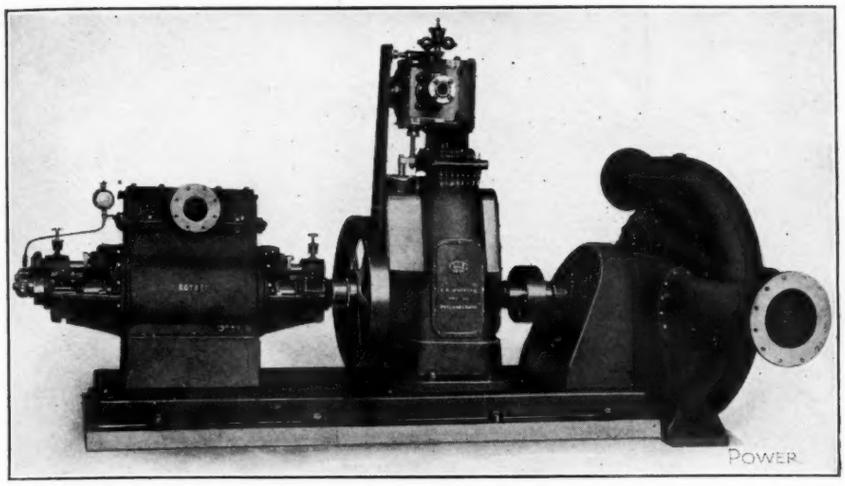


FIG. 4. A COMBINATION UNIT CONSISTING OF A VACUUM AND A CIRCULATING PUMP DRIVEN BY THE SAME ENGINE

ments this company has produced however, is what is termed the Rotrex combination system, this consisting of the Rotrex vacuum pump (which has previously been illustrated and described in POWER), combined with a centrifugal circulating pump, both directly connected to one vertical or horizontal engine.

Such a combination with a vertical engine is shown in Fig. 4. For units of medium size this makes a simple and desirable outfit. On the larger units, say above 2000 kw., the writer was told that it is more usual to operate the pumps separately.

Fig. 5 shows a complete installation of steam turbo-generator with surface condenser and Rotrex combination air and circulating pumps, engine driven. In this case the condenser is located directly under the turbine. The exhaust connection includes a copper expansion joint and a special type of gate valve made by the company, which has a side outlet for connecting the atmospheric relief valve.

This gate valve, shown in Fig. 6, consists of a regular gate valve with a wedge device for seating. The unusual feature is the provision of an outlet nozzle cast on the bottom of the valve body. In normal operation, with the gate open, steam passes directly from the turbine to the condenser, and, should the vacuum be lost, the automatic relief valve opens and the steam passes to atmosphere. If the condenser requires attention, necessitating shutting it down while the turbine must be operated (as is sometimes the case where there is an insufficiency of standby equipment), the gate valve can be closed, cutting out the condenser, but still permitting the steam to pass to the atmosphere through the automatic relief valve. This arrangement dispenses with the usual tee in the exhaust line which, with the large size of turbine exhaust connections, takes up considerable space, is very expensive and introduces another large bolted joint with its potential possibilities of air leakage. Provision is made for the water sealing of the stuffing boxes of the gate valve and the water sealing of the atmospheric relief valve, so as to avoid possibility of leakage at these points.

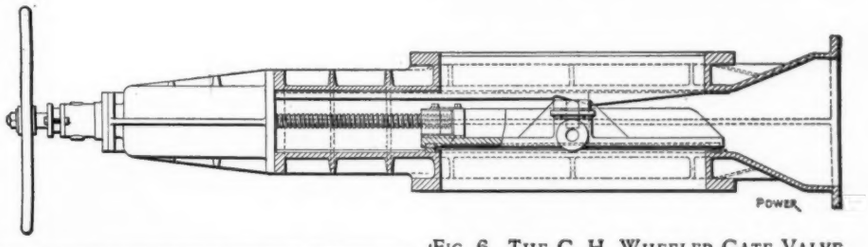


FIG. 6. THE C. H. WHEELER GATE VALVE WITH WEDGE SEATING DEVICE

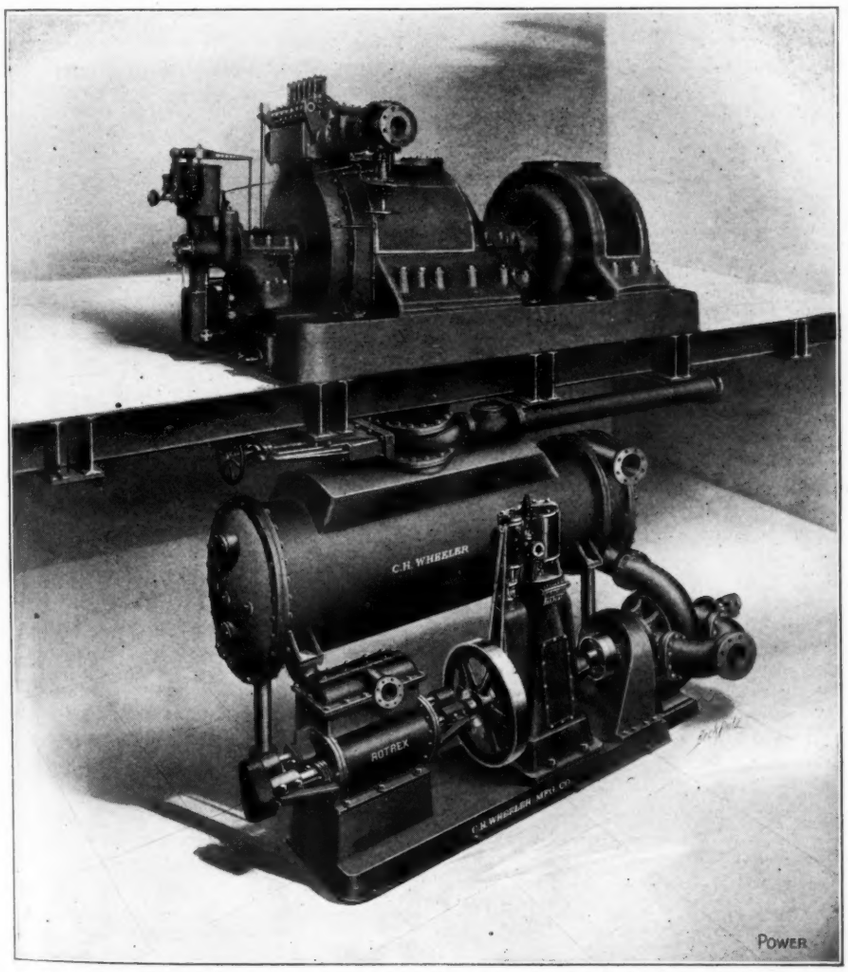



FIG. 5. SURFACE CONDENSER WITH ROTREX VACUUM PUMP AND CENTRIFUGAL CIRCULATING PUMP OPERATING WITH STEAM TURBO-GENERATOR

## SCHOOL NOTES

Two new scholarships in agricultural engineering, the first of their kind in any institution, have just been established at the Iowa State College. One was founded by the M. Rumley Co., La Porte, Ind., and is known as the Meinrad Rumley scholarship in honor of that pioneer manufacturer of agricultural implements. The

other was established by the International Harvester Co., of Chicago. Both are worth $250 a year to the students and will be awarded annually to young men qualified to do special research work along the lines of agricultural engineering. Prof. J. B. Davidson is head of the new department.

The Massachusetts Institute of Technology opened its summer surveying camp at Gardner Lake, East Machias, Maine, Aug. 7.

The camp forms part of the summer school maintained by the institute, and students in certain courses are required to attend the camp. Nearly 350 is the total registration this year, an advance over average figures even without taking into account the 70 who will be in the camp. An anonymous graduate bought the camp site and was planning for the buildings when another alumnus, C. W. Eaton, of Haverhill, offered $10,000 for buildings and equipment. The total plant has been established at a cost of about $50,000. There are three permanent buildings, living room, dining room and drafting room, about which are arranged 40 tents. There is an equipment of boats, motor boat and barge, and provision for bathing. Seventy students and twelve of the teachers, including practically the whole instructing force of the department with Prof. C. M. Spofford, will remain till the middle of September. The location of the camp gives unusual advantages for all the varieties of surveying.

# OBITUARY

### James J. Waters

James J. Waters, of Brooklyn, N. Y., a well known marine engineer, died on Aug. 1, from complications and shock resulting from an automobile accident.

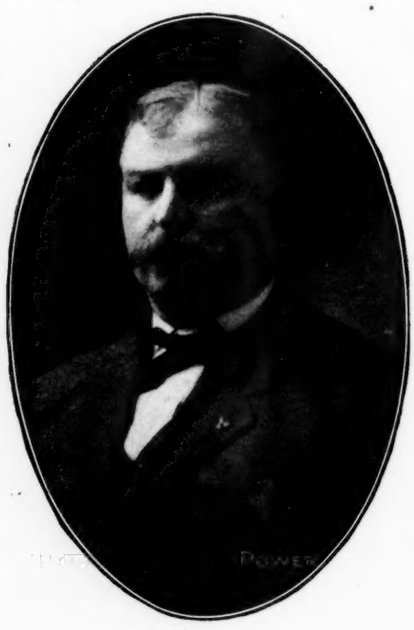
Mr. Waters was born in Jersey City, N. J., July 8, 1870. After serving his apprenticeship as a machinist he went to sea as engineer for the Morgan and Clyde Lines.

He was on watch in the engine room of the Old Dominion S.S. "Wyanoke" on the night of Apr. 29, 1896, when she collided with the U. S. S. "Columbia" in Hampton Roads, and was the only man in the engineers' department on watch at the time who escaped with his life. He was, however, so severely scalded that he was incapacitated for fourteen months.

After his recovery he served on harbor boats of the Old Dominion Line until 1903, when he was elected secretary of the Marine Engineers' Benevolent Association No. 33. He afterward became business manager, corresponding and financial secretary, serving successfully until his death. He was elected representative to the thirty-first annual convention in 1906 and has been reëlected annually ever since. In 1907 he was elected secretary and treasurer of the Board of Trustees of the *American Marine Engineer*. In January, 1911, he was elected chairman of the board and in January, 1912, was reëlected for a term of five years.

In the various association positions in which he has served he rendered exceptional service. He took a deep personal as well as official interest in all association matters and was well known and much loved in engineering and other circles.

Mr. Waters was the oldest son of John Waters, chief boiler maker for the Clyde Lines. He is survived by his widow, two children, father, mother, four sisters and a brother.



James J. Waters

He was an enthusiastic member of Jersey City Lodge No. 211, B. P. O. E., and of Fairview Council No. 498, Royal Arcanum.

### William Boissy

William Boissy, engineer of the power plant of the Woonsocket Rubber Co., Woonsocket, R. I., died recently at his home in that city. He had been seriously ill for some time.

### Harold H. Fletcher

Harold H. Fletcher, for many years a mechanical engineer for the Boston & Albany R.R., died at his home, 5 Winchester St., Brookline, Mass., on July 25. Mr. Fletcher had been employed by several large railroad interests previous to his connection with the Boston & Albany. He was a graduate of the Massachusetts Institute of Technology.

### Ralph P. Gillette

The death is announced of Ralph P. Gillette, manager of mechanical sales for the Minneapolis Steel & Machinery Co., Minneapolis, Minn., on July 12. Mr. Gillette had sought shelter under a tree during a severe storm and was instantly killed by lightning.

### Leland B. Smith

Leland B. Smith, once chief engineer of the Yazoo Light & Power Co., Yazoo City, Miss., and later with the Georgia Power Co., Atlanta, Ga., recently met his death by drowning.

Mr. Smith was about 25 years of age, and his rapid advancement in the engineering field gave promise of his becoming one of the ablest of the young engineers in the South. He was instrumental in organizing Col. Goethals Branch No. 1, I. O. E., and was its first secretary.

# PERSONAL

Bartholomew Scannel, of Lowell, Mass., who has represented the boiler-manufacturing interests of the Massachusetts State Board of Boiler Rules for several years, has declined a reappointment to that office. Gov. Eugene N. Foss has named Henry H. Lynch, of Brookline, in his stead.

Lieut. Walter G. Diman, son of George H. Diman, of the American Woolen Co., at Lawrence, Mass., has been appointed superintendent of motive power for the Amoskeag Mills, Manchester, N. H. Capt. Charles H. Manning, who has so long been identified with this plant, will retain his connection with the corporation.

Alfred Still was recently appointed chief electrical engineer of the Algoma Steel Corporation, with headquarters at Magpie Mine, Ont. He is a member of the British Institution of Electrical Engineers, the Institution of Civil Engineers and the American Institute of Electrical Engineers, and is the author of standard electrical books and a contributor to technical journals.

President C. A. Griscom, Jr., and Vice-President Arvine Wales are the guiding officials of the Griscom-Russell Co., which has taken over the business of the Griscom-Spencer Co. and the Russell Engine Co., of Massillon, Ohio. The new company will assume all the obligations of both companies, and carry on all the business heretofore conducted by them. So far as is necessary, pending business will be completed by the former companies. The management and staff remain unchanged. The general offices of the Griscom-Russell Co. will be at 90 West St., New York City.